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THE DEMING COMPANY
(ESTABLISHED 1880.)
SALEM, OHIO, U.S.A.

103/1.3

CENTRIFUGAL PUMPS - THEIR THEORY AND APPLICATION

Letter No. 1

415
Digest 6

GENERAL DISCUSSION OF PUMPS

415
Digest 6

A pump, broadly speaking, is a mechanical device for elevating a liquid from one level to another, or for transporting it a certain distance, or for both; or for receiving the liquid under pressure and discharging it against a higher pressure.

There are two general classes of pumps - those worked by hand and those that are operated by some form of mechanical power, such as steam engine, steam turbine, electric motor or gasoline engine. In this article we are concerned with Power Pumps only, and Centrifugal pumps in particular. Power pumps are divided generally into three classes:

(a) Reciprocating Pumps.

(b) Rotary Pumps.

(c) Centrifugal Pumps.

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(a) The reciprocating pump, as the name implies, is a device wherein the water is displaced by the action of reciprocating pistons or plungers.

(b) The rotary is another form of the displacement pump, but the displacing action is brought about by the rotation of two intermeshing cams which squeeze out the water between them.

(c) The centrifugal pump, as its name implies, works through the action of centrifugal force.

Strictly speaking a fourth type of pump should be added which has the outside appearance of a centrifugal pump, but the rotor is more of a bucket type and does not have centrifugal pump characteristics.

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The Centrifugal Pump may briefly be explained as follows:

If a bucket of water is held in the hand, and whirled rapidly around in a circle, the water will not pour out, but will remain in the bucket. It is held there by centrifugal force. If a hole is cut in the bottom of the bucket, the water will issue in a stream therefrom and the faster the bucket is rotated the farther the jet of water will project.

This is a rough illustration of the centrifugal pump.

The action of centrifugal force is manifested by the pull the bucket and the water exerted upon the holder's arm. The pull that has to be exerted to keep the bucket with the contents from flying out of the hand is equal and opposite to the pressure that the water exerts upon the bottom of the pail. Now if a pipe could be imagined connected to the bottom of the bucket and bent upward, this pressure would cause the water to rise in it, and if the rotative speed were sufficiently high, water would flow out of the upper end of the pipe.

This action, of course, would in time empty the bucket. Means must be provided for renewing the contents of the pail as fast as it is emptied. If some sort of supply pipe could be connected, we would have, in effect, a crude centrifugal pump.

In reality, the action is as follows: (See Figs. 1 & 2, also use for reference any of our centrifugal pump bulletins showing the complete cross-section and pictures). As soon as the pump casing and the suction pipe have been filled with water the pump can be started and the water in the impeller is thrown from the center outward, due to centrifugal force. This forces the water already in the casing out through the discharge pipe, thus producing discharge.

The water moving out of the impeller tends to produce a vacuum in the center. Atmospheric pressure, acting upon the surface of the source of supply, immediately forces more water up the suction pipe and into the impeller to replace that which has been thrown out due to centrifugal force, and we often speak of this action as the pump lifting or sucking the water. The action is steady and continous, hence the pump functions.

Types of Centrifugal Pumps.

There are many types of centrifugal pumps and the various text books and manufacturer's catalogs give an unlimited number of cuts and illustrations.

Classification of centrifugal pumps is made according to the TYPE and NUMBER of impellers used, and both classes can be built as Horizontal or Vertical pumps, i.e. with shaft located in a horzontal or vertical plane.

(a) Side suction or single inlet, where the water enters the center of the impeller from one side only. (Figs. 1, 2 and 3)

(b) Double suction or double inlet pumps, where the water enters the center of the impeller from both sides. (Fig. 4)

Both Side Suction as well as Double Suction inlet impellers can be made of the open or enclosed type, i.e. the channels through which the water flows while passing through the impeller are open on the side of the first type, (Fig. 1), and are fully enclosed on the second type, (Fig. 3).

Impellers are called many names, the most common ones in addition to impeller being: Runners, rotors, paddlewheels, pumpers, fans, pistons, etc.

The open type impeller (Fig. 7) consists of a hub with a number of arms or wings or vanes fastened thereto pointing outwardly. These wings may be straight or curved, the shape, of course, having a great effect upon the performance of delivering water. In order to force the water from the inlet or center of the impeller towards the outside, also called periphery, this open type impeller has to rotate between stationary side walls which prevent the water from being thrown out towards the sides. (See Fig. 1 and Fig. 4, upper half.)

The enclosed impeller (Fig. 3) has the same hub and vanes as the open type impeller but, instead of rotating between stationary side walls, these side walls or shrouds are cast on the impeller and rotate with it, the water then passing from inlet of impeller to the outlet through enclosed passages or so-called channels. (Fig. 3 and Fig. 4, lower half.)

In a later letter we will discuss at length advantages and disadvantages of both the enclosed and open type impellers.

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The various types of pumps are classified according to the impeller.

Figs. 1 and 2 show cross sections of a side suction pump with an open type impeller.

Fig. 3 shows the same side suction pump with enclosed type of impeller.

Fig. 4 (upper half) shows a double suction pump with an open type impeller. Note that the water enters through the suction opening of pump and then immediately is divided into two streams, one going toward the left, the other going towards the right, and when both streams strike the shaft they are turned again towards the impeller and enter same on both sides of it.

Fig. 4 (lower half) shows a double suction pump with an enclosed type impeller.

Fig. 5 shows a so-called multi-stage centrifugal pump.

When the head to be pumped against is too great for a single stage pump, a two, three, four or more stage (multi-stage) pump is used.

This consists virtually of two or more pumps in one casing, and connected in series. That is, the first stage or impeller discharges into the second, the second into the third, and so on. The pressure increases in direct proportion to the number of stages. If we have a five-stage pump and each stage is designed for fifty pounds pressure, the pressure at the final discharge will be two hundred fifty pounds.

The effect is just the same as though five separate pumps were piped together so that the discharge of the first entered the suction of the second; the discharge of the second went to the suction of the third, and so on. Placing the impellers on one shaft in a common casing reduces the space occupied, and cuts down the number of bearings and stuffing boxes.

Multi-stage pumps are built without guide vanes (Fig. 5), and with guide vanes (Fig. 6), according to the preference of the manufacturer and, sometimes, specifications of the customer.

We find very few multi-stage pumps at the present time which are built with an open type impeller. Although we do not see any reason why good results could not be obtained with open type impellers when moderate heads per stage are being used. A fair illustration is the so-called Deep Well Turbine Pump where quite a few manufacturers are using the open type impeller.

Fig. 6 shows another multi-stage pump which is of the double suction impeller type. This construction is more cumbersome on account of complicated core work of the casing, but a few manufacturers prefer it. We cannot see any special advantage except for very large quantities at high speeds, and we will later on, when discussing the impellers in general, return to this type of pump.

In order to simplify the casing construction as much as possible, frequently the Diffusor is used. This Diffusor or guide vane is an apparatus which consists of two circular plates, or walls held together by a species of ribs called vanes. Channels are formed between these vanes and the sidewalls through which the water from the impeller is discharged. These channels are of gradually increasing cross-sectional area, and the water, in passing through them, is gradually slowed down to the desired velocity in the casing proper. Fig. 6 shows a special diffusor which has, in addition to the radial passages, lateral passages which let the water pass over to the opposite inlet side of the impeller but this construction is at the present time of little importance to us and, therefore, we abstain from discussing it at length.

A few additional remarks regarding the diffusor, or guide vanes, however, we believe are in order at this time. Pumps having such diffusors or guide vanes are sometimes called "Turbine Pumps" on account of their similarity to the Water Turbine or Water Wheel, being the reversal of the same.

In a water turbine the wheel is surrounded by a stationary ring with vanes guiding the water into the wheel, finally discharging through the center into the tail stock. In a centrifugal pump the water follows the reverse course, namely, from the inside to the outside, and any pump which has guide vanes around the outside of the impeller is called a "Turbine Pump".

The first multi-stage pumps were all of the diffusor type and were then called Turbine Pumps.

The tendency, however, during the last ten years has been to do away with diffusors and today most multi-stage pumps are built without diffusors.

Now the Centrifugal Deep Well Pump is pretty well monopolizing the term "Turbine Pump". The fundamental laws of a high-grade diffusor are, however, in most designs sadly neglected, but, never-the-less, vanes and guide channels around the impeller are present, serving as a means to tie the bearing housing to the wall of the bowl and provide water passages.

Figs. 1 to 6 inclusive all illustrate centrifugal pumps of the horizontal construction, i.e. the shaft is in a horizontal position.

Centrifugal pumps as mentioned above are also made with shafts in a vertical position and we have vertical pumps of all types similar to the horizontal pumps.

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A comparatively few years ago the centrifugal pump was practically unknown. Within the past twenty years such strides in its design and manufacture have been made that it is now one of the principal types of pumping machinery. Originally regarded as of low efficiency, and useful only for large volumes against low heads, it is now used for small and moderate capacities also, and for heads up to 3000 ft. or more. The efficiency has been improved until, under favorable conditions, it equals that of the reciprocating pump. Of late, efficiencies well over 80 percent have been obtained in commercial designs.

During the first six months of 1930 approximately 50,000 centrifugal pumps were marketed in the U.S.A. While the centrifugal pump is gaining in favor daily, and gradually supplanting most reciprocating pumps, it is not suitable for every pumping service, and must be intelligently selected and applied. There are some services that will probably always belong to the reciprocating pump.

The principal advantages of the centrifugal pump are:

(1) Simplicity.
(2) Compactness.
(3) Low initial cost.
(4) Low maintenance cost.
(5) Lack of reciprocating parts.
(6) Good efficiency.
(7) Adaptability to high speed prime movers.

On the other side of the ledger must be set down as disadvantages:

(1) Lack of suction power (the centrifugal must always be primed, to operate on a suction lift).
(2) Sensitiveness to variations in head or speed.
(3) A wide-spread ignorance of its characteristics.

This last, of course, cannot be blamed upon the pump. The benefits of the educational work of the last fifteen or twenty years are more and more apparent, and consequently the troubles arising from ignorance of this type of pump are lessening year by year.

We hope and are confident that this course will go far toward bringing about general familiarity with the centrifugal pump.

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Salem, O.

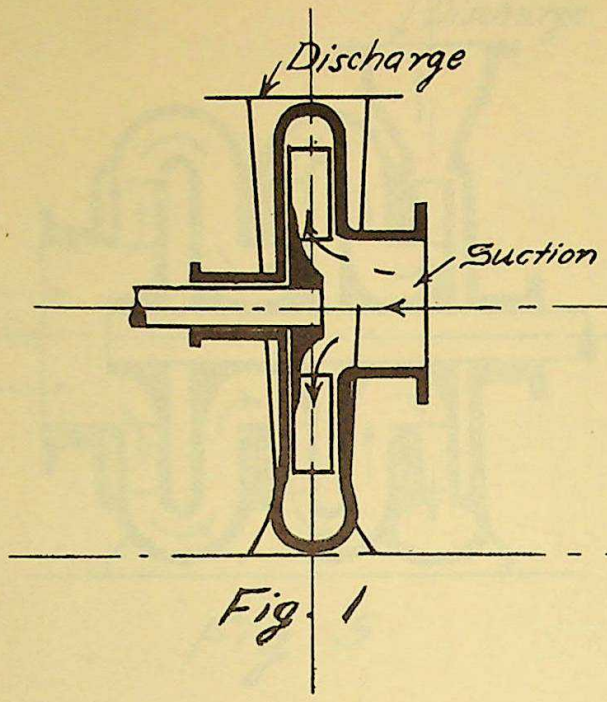

Fig. 1

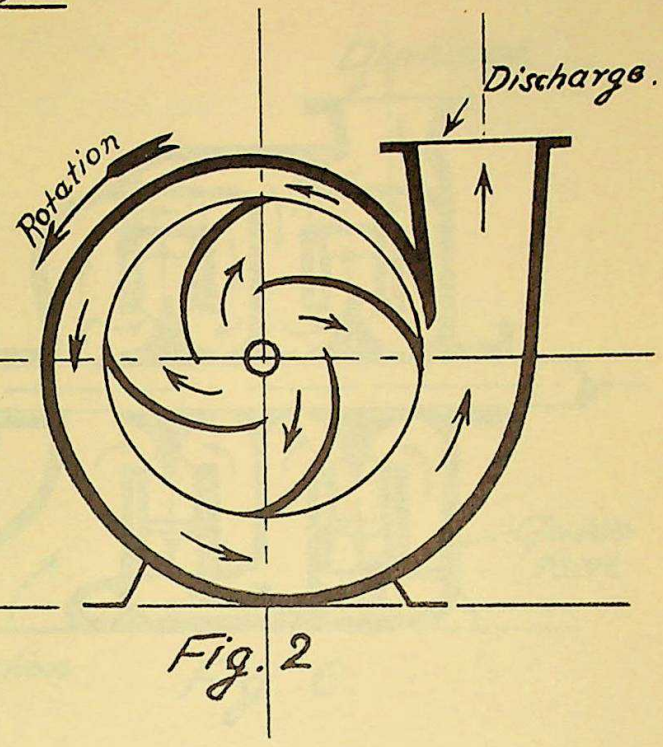

Fig. 2

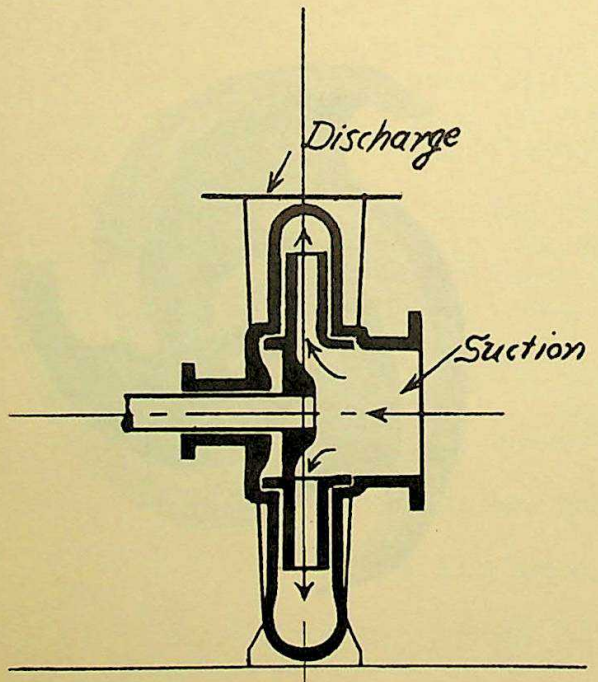

Fig. 3

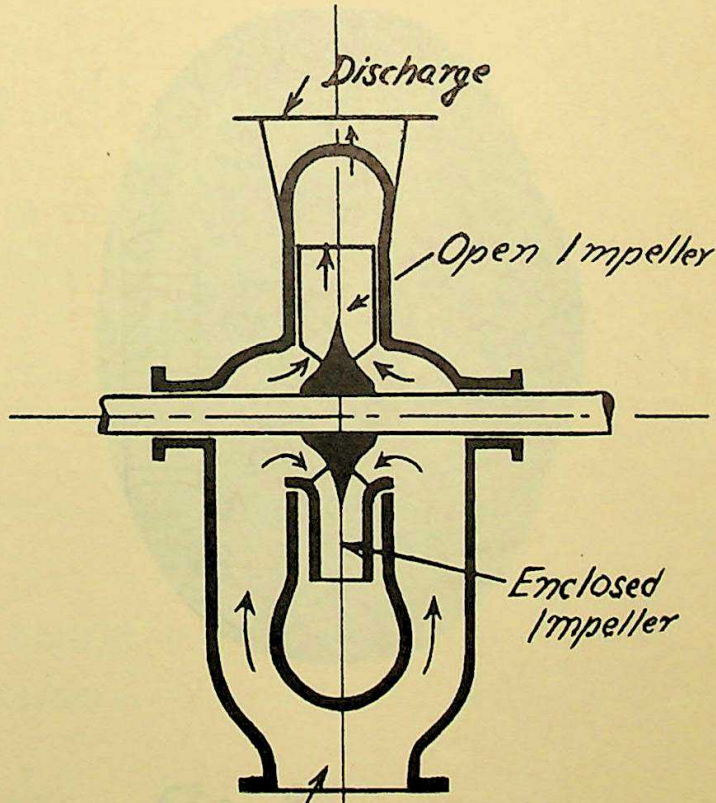

Fig. 4

Plate #1.

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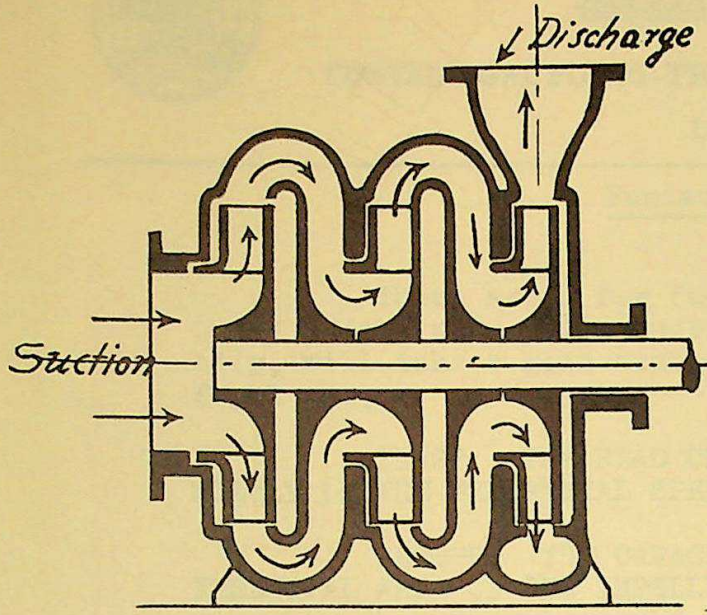

Fig. 5

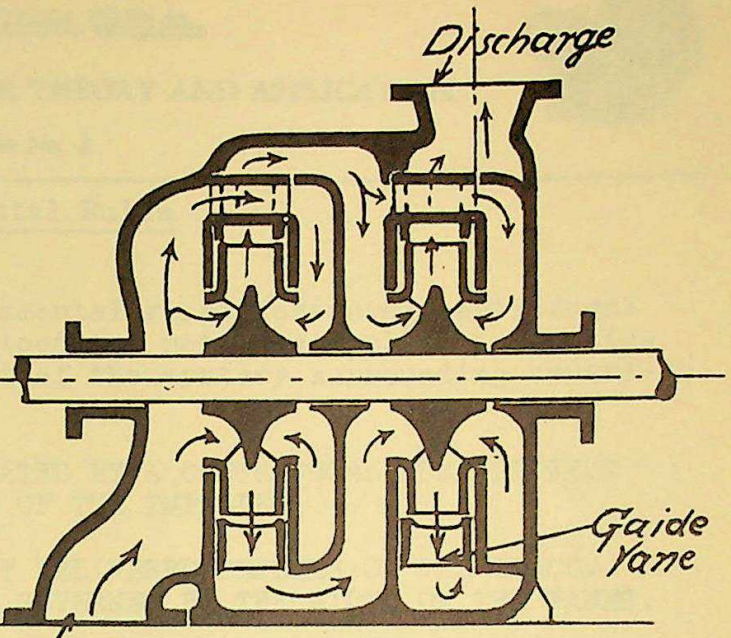

Fig. 6.

Fig. 7

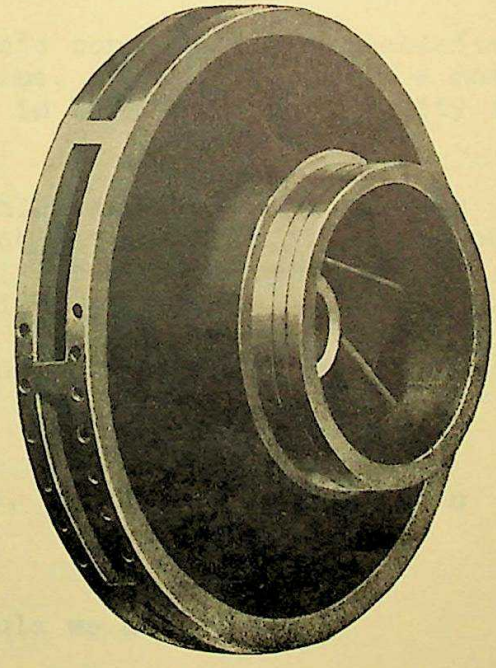

Fig. 8

Plate #2

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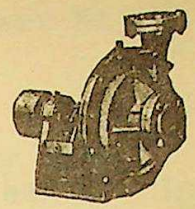

CENTRIFUGAL PUMPS - THEIR THEORY AND APPLICATION

Letter No. 2

Fundamental Rules

There are a few fundamental rules governing centrifugal pumps which, when fully understood and not only applied automatically, will take a good deal out of the mystery surrounding centrifugal pumps.

FIRST: THE HEAD CREATED BY A CENTRIFUGAL PUMP DEPENDS SOLELY ON THE PERIPHERAL SPEED OF THE IMPELLER.

SECOND: THE CAPACITY DELIVERED DEPENDS ON THE CIRCUMFERENTIAL AREA OF THE IMPELLER GOVERNED BY THE WIDTH OF THE VANES.

Note: The designer, of course, has means and ways to vary these rules within certain limits.

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Velocity

Velocity is one of the basic conceptions of mechanics and is defined as distance divided by time. For instance, if a motor car travels a distance of 300 miles in 10 hours, its velocity is 30 miles per hour on the average.

Velocity is the rate at which an object changes its position with reference to another object. In the illustration just given, the car changes its position with respect to the earth's surface at the rate of 30 miles per hour.

The most common expressions, for velocity are:

Feet per second,	or	ft/sec.
Revolutions per minute,	"	R.P.M. or n
Miles per hour,	"	m.p.h.

Putting it down as a formula we have:

Velocity (v) = Distance (s) divided by time (t)

$$v = \frac{s}{t}$$

Velocity is either absolute or relative. In most cases it is absolute, as in the motor car illustration, but in centrifugal pump design, there are several velocities that are relative, as will be explained later.

The absolute velocity of a body is its velocity with reference to a fixed object standing still, such as the earth. In mechanics, we consider the earth as a stationary, fixed body.

The relative velocity of a body is its velocity with reference to a moving object, such as a man walking down the aisle of a railway train in motion, that is, his velocity with respect to the train, and not to the earth.

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The peripheral speed of an impeller can be compared with the rim speed of a pulley. Such rim speed expresses the distance of a point at the periphery, or outside rim, travelled through in a certain length of time.

The circumference of any circle is

$$\text{Diameter} \times \pi$$

where π is a constant which is equal to 3.14. If a pulley of 12" diameter turns around once, each point of the rim has travelled

$$12" \times 3.14 = 37.68", \text{ or expressed in feet}$$

$$1' \times 3.14 = 3.14'$$

If the pulley turns around 1750 times, the distance covered by each point of the periphery is

$$3.14 \times 1750 \text{ or } 5495'$$

The turning around we call revolutions, and by entering the time element we have the usual term of revolutions per minute, R.P.M., as mentioned above. Putting this in the form of a formula we have

Rim speed, or circumferential speed, = Diameter of impeller (pulley) $\times \pi \times$ R.P.M.

Velocities in centrifugal pump calculations are mostly expressed in feet per second, and impeller dimensions are usually given in inches (1' = 12"); therefore, the above equation has to be divided by 12 and also by 60 (1 min. = 60 sec.) and the following formula results:

$$\text{Circumferential speed} = \frac{\text{Dia.} \times \pi \times \text{R.P.M.}}{60 \times 12} = \frac{\text{Dia.} \times \pi \times \text{R.P.M.}}{720}$$

The circumferential speed depends on two variables, the diameter of the impeller and the revolutions. It is easily seen that an impeller of 6" diameter turning at 3500 R.P.M. will have the same circumferential speed as a 12" impeller at 1750 R.P.M.--double the diameter at half the speed.

Proof: $$\frac{6" \pi 3500}{720} = 91.5 \text{ ft/sec.}$$

or: $$\frac{12" \pi 1750}{720} = 91.5 \text{ ft/sec.}$$

Example:

How large will the impeller diameter be when running at 1150 R.P.M. and to have the same circumferential speed of 91.5 ft/sec. as above?

$$\text{New Impeller Diameter} = \frac{91.5 \times 720}{1150 \times \pi} = 18.25"$$

The above given first fundamental rule stated "The head created by a centrifugal pump depends solely on the peripheral speed of the impeller", and the 91.5 ft/sec. circumferential speed is equivalent to a certain head which an impeller will create, no matter if this velocity is obtained with a 6", 12" or 18¼" diameter impeller and the above given corresponding speeds, or with a 3" impeller turning at 7000 R.P.M.

High speeds are here to stay. Twenty-five years ago many engineers objected to 1750 R.P.M., and even 1150 R.P.M. Today we have the same hesitation about 3500 R.P.M.--"The pump runs too fast, the water has excessive velocities in the casing", etc. For a certain head the velocities through the pump are the same, no matter if the pump runs at 100 or 10,000 R.P.M. Of course, if velocities within the pump become excessive, multi-stage pumps are being used; to be discussed later on.

The only advantage of a slow speed over a high speed pump can be found in less rubbing speed where the shaft passes through the stuffing box, but a high speed pump is usually equipped with a smaller diameter shaft for the corresponding horsepower to be transmitted, and with a high grade stuffing box packing the rubbing speed in the stuffing box can well be forgotten when clean water is being pumped.

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A few fundamentals, physical and mechanical, and consequently hydraulic laws, must now be discussed. They are being explained in a simple manner and can be understood by everybody, and real understanding of the same, instead of mere automatic application of these proven facts, will be of great help.

We immediately go "in medias res"--in the midst of things--as the old Romans used to say, and give the formula, with which every hydraulic engineer is most familiar:

$$h = \frac{v^2}{2g} \qquad v^2 = 2gh \qquad v = \sqrt{2gh} \qquad (1)$$

or its transposition

$$v = \sqrt{2gh} \qquad (2)$$

The letter g represents the acceleration of gravity; h is the head in feet, and v represents velocity in feet per second.

Sir Isaac Newton, the English philosopher and scientist, was the first to establish the basic laws of the universe and also the law of terrestrial gravitation which will be discussed later on.

After many discouraging and trying attempts to have his theories substantiated by results he finally succeeded, and the laws he established are still true after a period of over two and three quarter centuries.

Prof. Einstein's relativity may change things a bit, but Sir Isaac Newton's law of gravitation will serve our purpose--and we can grasp it.

---o---

Law of Attraction

The "Force of Gravity" is a term which denotes the attraction between the earth and its bodies. The direction of this force never changes, it always acts in a straight line from the center of the body (center of gravity) towards the center of the earth.

If you hold a body, a stone for instance, you will notice a downward pull and if you release your hold on the stone it will fall to the ground.

Such downward pull while called Weight is simply the attraction between the earth and the body.

(a) Bodies weigh most at the surface of the earth.

(b) Below the earth's surface the weight decreases as the distance to the earth's center decreases.

(c) Above the surface the weight decreases as the square of the distance from the earth's center increases.

The mean diameter of the earth is considered at sea level and all measurements, such as elevations and depressions etc. are referred to sea level; hence, when we speak of the surface of the earth we mean sea level.

The earth's radius in round figures is 4000 miles.

First Illustration:

How much does a body which weighs a 1000 pounds at the earth's surface, weigh when 2000 miles from the earth's center?

As per (b) we have the direct proportion

Present weight to new weight as present distance to new distance

$$1000 : w' = 4000 : 2000$$

$$w' = 500 \text{ lbs.}$$

At the center of the earth (no miles) it should weigh nothing, since, the attraction of gravity decreases from a maximum at the surface, to zero at the center of the earth.

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Second Illustration:

A flying machine weighs, for instance, 3000 lbs. at the earth's surface (4000 miles from the center of the earth). The weight of this machine 6000 ft. or 1.135 miles in the air would be:

Present weight to new weight as new distance to present distance

$$3000 : w' = 4001.135^2 : 4000^2$$

$$w' = 2998 \text{ lbs.}$$

---o---

In accordance with the foregoing, we can write the following formulae:

For the weight of a body below the earth's surface

$$w : w' = r : r'$$

For the weight of a body above the earth's surface

$$w : w' = r'^2 : r^2$$

---o---

Weight and Mass

As stated previously "Weight is the downward pull due to the attraction between the earth and the body". This attraction is the "Force of gravity".

If the weight of the body is divided by the acceleration of gravity, we call the result "Mass."

Mass is an intangible thing called by some authors "Quantity of Matter" and is used in many psysical and engineering calculations.

It is proportional to weight, in fact equals weight divided by g.

$$m = \frac{w}{g}$$

It has been stated that the weight of a body varies in relation to sea level. This is because Gravity varies.

Mass never changes.

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In the second illustration of the previous topic, we stated that a flying machine weighing 3000 lbs. at the earth's surface weighed but 2998 lbs. at a distance of 6000 ft above the earth.

The mass of the aeroplane has undergone no change, but the value of g has decreased, as this elevation is farther away from the earth's center than the sea level.

It would be impossible to prove this with the ordinary weight measuring instruments. The best experiment to prove this law could be conducted with a spring balance at the top of a high mountain.

The spring balance measures weight by the amount of the elongation of its spring, whereas the scales measure weight by comparing the mass of the object to be weighed with the mass of what is called the "Standard Weight" which is a standard of comparison. If the scales were used on the mountain top, no change in weight would be observed, as the respective masses of the object to be weighed, and the standard of comparison, would have undergone no change and would still balance.

The spring balance on the other hand is extended by a given force which is the weight of the object, hung upon it. If the weight is taken and recorded at the base of the mountain and taken again at the summit, it will be found that the amount of elongation of the spring is perceptibly less. The weight in each case is the product of the mass of the body, which is constant, and the attraction of gravity. As g is less on the summit than at the mountain base, the product is less, or the weight w is less, accounting for the difference in elongation of the spring of the balance.

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As above stated, some authorities define mass as quantity of matter.

Thus a cubic foot of water weighs 62.4 lbs. at 62° F. which means that the mass of water is such that a volume of one cubic foot of it is drawn towards the earth's center with a force of 62.4 lbs.

The mass of different substances is not the same, for instance, while the cubic foot of water weighs 62.4 lbs., a cubic foot of alcohol weighs but 49 lbs.

There is approximately 1.275 as much matter in the cubic foot of water as in the cubic foot of alcohol, or conversely, the quantity of matter in .785 cubic foot of water equals the quantity of matter in 1 cubic foot of alcohol. This brings us directly to the term "Specific Gravity."

---o---

Specific Gravity

The unit of measurement is 1 cubic foot of water at 62° F. which weighs 62.4 lbs and its "Specific Gravity" is "1".

The specific gravity of all other substances, solids, liquids, or gases, is "The ratio of the weight of a given volume of such substances to the weight of an equal volume of water."

First Illustration:

As stated above, 1 cubic foot of alcohol weighed 49 lbs. The specific gravity of the alcohol is the ratio of 49 to 62.4, or

$$\frac{49}{62.4} = .785 = \gamma = \text{Specific Gravity}$$

Second Illustration:

A gallon of brine weighs 10.35 lbs. and a gallon of water 8.33 lbs. The specific gravity of the brine is

$$\gamma = \frac{10.35}{8.33} = 1.25$$

We, therefore, have the formula

$$\frac{\text{Weight of Substance}}{\text{Weight of same volume of water}} = \text{Specific Gravity} = \gamma \quad (6)$$

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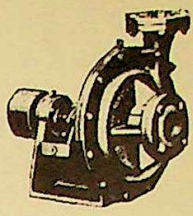

CENTRIFUGAL PUMPS - THEIR THEORY AND APPLICATION

Letter No. 3

Pressure

By reason of the attraction of gravitation one body resting upon another exerts a pressure upon it. For instance a standpipe full of water exerts a pressure upon its base, or upon the pipes and their contents leading from it into the city mains.

A steam boiler, owing to the expenditure of heat energy upon it is charged with a form of energy that tends to burst the shell, heads, and tubes.

Air, forced into an automobile pneumatic tire, keeps the tire inflated and exerts a bursting pressure against its inner walls.

All these forms of pressure, and others, are expressed as so many pounds per square inch, kilograms per square centimeter, or in feet of water, or in inches of mercury.

The atmosphere surrounding us exerts a certain pressure. This equals 14.7 pounds per square inch at sea level, and decreases with altitude or elevation above sea level.

If a hollow enclosed vessel should have the air entirely removed from it, it would be found that the external surrounding air would press inward upon the walls of the vessel with an intensity of 14.7 lbs. per square inch at sea level, tending to cause the vessel to collapse.

There are what we may term, two pressures, "gauge", and "absolute".

GAUGE pressure is that denoted by an instrument called a pressure gauge and is the pressure above atmosphere.

ABSOLUTE pressure is gauge pressure plus atmospheric pressure.

Consider a pressure gauge attached to the bottom of a standpipe. The pressure shown is merely due to the height of the water in the standpipe; it is due to the weight of a column of water 1" square. The atmospheric pressure acting upon the surface of the water at the top of the standpipe is balanced by the same atmospheric pressure acting upon the outlets. The pressure gauge, when reading zero, is already acted upon by the atmosphere. Therefore, the reading it shows when connected is due only to the pressure of the column of water in the standpipe, or merely shows pressure above atmosphere.

Referring to the other foregoing examples, boiler pressure and the pressure in the pneumatic tire, are gauge pressures.

When forcing water a pump produces a pressure which can be registered by a pressure gauge or mercury column. When sucking water, however, a pump tends to produce a vacuum, i. e., to reduce the atmospheric pressure within the pump so that the atmospheric pressure outside pressing upon the surface of the reservoir from which the water is taken, causes the water to rise and fill the pump. In this case the pump also works against a gauge pressure, although this pressure is below atmospheric pressure, and may be termed a negative pressure. It is registered by a vacuum gauge or by a mercury column, which will be described later.

A vacuum is a space in which the pressure is below atmosphere. It is the pressure existing in a closed vessel from which the air has been partially extracted. In a perfect vacuum, never obtained in practice, there would be no pressure at all.

Conversion Formulae

It has just been stated that atmospheric pressure (at sea level) equals 14.7 lbs. per square inch. It is also equal to a column of water 407.37 inches high, equalling 33.95 feet high, equalling 30" mercury high.

That is to say, 1 lb. per square inch equals a column of water 2.31 feet high. How is this ratio arrived at? We will try to demonstrate this by an easily understood example, also some other conversion constants.

A cubic foot of water weighs 62.4 lbs. Consider this composed of 144 columns, each 1" square or 1 square inch cross sectional area, and 1 foot high, there being 144 square inches in one square foot. The weight of each of these columns of water is, therefore, 62.4 ÷ 144 equals .433 lbs. That is, each column of water one foot high exerts a pressure at its base of .433 lbs.

Therefore, .433 lbs. per square inch pressure is equivalent to a column of water 1 foot high. Accordingly a pressure of one lb. per square inch equals a column of water 1 ÷ .433 ft. high, equals 2.309 feet high, or 2.31 feet in round numbers. Therefore, in order to convert pressure in pounds per square inch to equivalent feet of water.

Multiply Pounds by 2.31

(7)

or $h = 2.31p$ (Head = 2.31 times pressure)

This formula is used constantly in pump work, as a pump is usually designed to work against a head in feet rather than pounds pressure. A centrifugal pump in particular has to be calculated from the head in feet. The velocity of the impeller depends on the formula described in letter #2, sheet #4.

$$v = \sqrt{2 g h} \qquad (7)$$

h is head in feet, and not pounds pressure. If the specification states that the pump is to produce so many pounds pressure, this quantity must be converted to feet before the pump can be designed.

Furthermore, as work is the product of weight, or resistance, times distance, pounds pressure in order to represent distance or height, must be converted into feet in order to calculate the power needed to operate the pump.

---o---

Another conversion formula much used in pump work is

1" of mercury = 1.132 ft. of water. (8)

We will explain this as follows, using the same "dead weight" method as in the foregoing explanation of conversion from pounds pressure to feet of water.

A cubic foot of mercury weighs 13.6 times as much as a cubic foot of water, or about 849 lbs. Consider it made up of 144 columns, each 1 ft. or 12" high and of one square inch cross sectional area.

The weight of each column is therefore

$$\frac{849}{144} = 5.88 \text{ lbs}$$

Or each column exerts a pressure of 5.88 lbs., and as the area at the base of each column is 1 sq. inch, this pressure is 5.88 lbs. per square inch.

A column 1" high would exert 1/12 of this pressure, or .49 lbs. per sq. inch.

But we have already shown that 1 lb. per square inch is equivalent to 2.31 ft. of water. Therefore, .49 lbs. equals .49 x 2.31 equals 1.132 ft. of water.

Or 1" of mercury equals 1.132 ft. of water.

---o---

It may be wondered why mercury is used in gauges, as above described. Mercury is the heaviest of all the liquids. Any other liquid could be used as a medium of measurements, water for instance, but an exceedingly bulky gauge would result. One pound pressure, for instance, equals a column of mercury only 2.0435" in height, while the equivalent height of a column of water is 2.31 feet. 100 pounds pressure, therefore, which is very common in pump practice, would have to be measured by a water column 231 ft. high, which is obviously impractical.

The equivalent height of a column of liquid to correspond to a given pressure varies inversely as the specific gravity of the liquid. That is, the heavier the liquid, the lower the column and vice versa. Therefore, the minimum height of the colum will correspond to the heaviest liquid, Mercury, which results in the smallest and most compact liquid gauge.

---o---

Second Fundamental Rule

In order to advance further in the understanding of a centrifugal pump and not to become too tiresome with explanations of mechanics and hydraulics, which groundwork, however, is very essential, the following statement of letter #2 is now taken up again:

"The capacity depends on the circumferential area of the impeller governed by the width of the vanes."

---o---

Quantity or Volume Discharged

If f is the area of an orifice, then the volume flowin through the same will be, per second,

$$Q = f\ v \qquad (9)$$

Since v equals velocity in feet per second, and since velocity taking place across a certain area results in volume, if f is expressed in square feet then Q is the volume in cubic fee per second.

If t represents the number of seconds, then

$$Q = f\ v\ t$$

represents the total volume delivered in t seconds.

---o---

We usually figure areas in square inches, velocities in feet per second and volumes in gallons per minute. This changes the formula to

$$Q = \frac{f\ v}{.321} \qquad (10)$$

where Q is expressed in gallons per minute.

Formulae (9) and (10) give the theoretical volume that should be discharged, and the transformation of this formula from (9) into (10) should be accepted as correct, its derivation being simply mechanical.

Illustration

A pump is required to have a capacity of 100 G.P.M. The impeller is 6" diameter operating at 1750 R.P.M. What is the width of the impeller at its periphery?

$$f = \frac{.321 \times Q}{v}$$

Inserting the quantity required, we have

$$f = \frac{.321 \times 100}{v}$$

The area f at the outside diameter of the impeller is represented by formula (1) multiplied by the width (b) of the impeller, Fig. 1, plate 1 and we have

$$f = D \times \pi \times b$$

Using this new value of f in the previous equation, we have

$$D \times \pi \times b = \frac{.321 \times 100}{v}$$

v in our case represents the velocity of the water through the impeller. Refer to Fig. 2, plate 1 and imagine that the water travels from the center to the outside in a strictly radial direction and the orifice or area through which the water leaves the impeller is the above

$$f = D \times \pi \times b$$

From this the area of the four blades which is taken up by them the outside should be subtracted, but this is just now of minor importance.

Let us assume $v = 10$ ft./sec.

We then have

$$D \times \pi \times b = \frac{.321 \times 100}{10}$$

Solved for b

$$b = \frac{.321 \times 100}{D \times \pi \times 10}$$

For a 6" diameter impeller

$$b = \frac{.321 \times 100}{6 \times \pi \times 10} = .545"$$

The impeller is .545" or approximately 35/64" wide at the tip.

It will be noted that the R.P.M. at which the impeller rotates did not enter into the calculations regarding its width.

The only selection to be made is the radial velocity through the impeller. Of course, it is very important to choose this value correctly in relation to the head against which the pump has to operate and also with relation to the exit angle of the impeller vanes. A wrong value might still give the quantity desired but the efficiency would be greatly reduced. As approximate values the radial velocity may be assumed with

6 to 9 ft for heads up to 50 ft
10 " " " " " 100 "
11 " " " " " 150 "
12 & higher for higher heads

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SALEM OHIO.

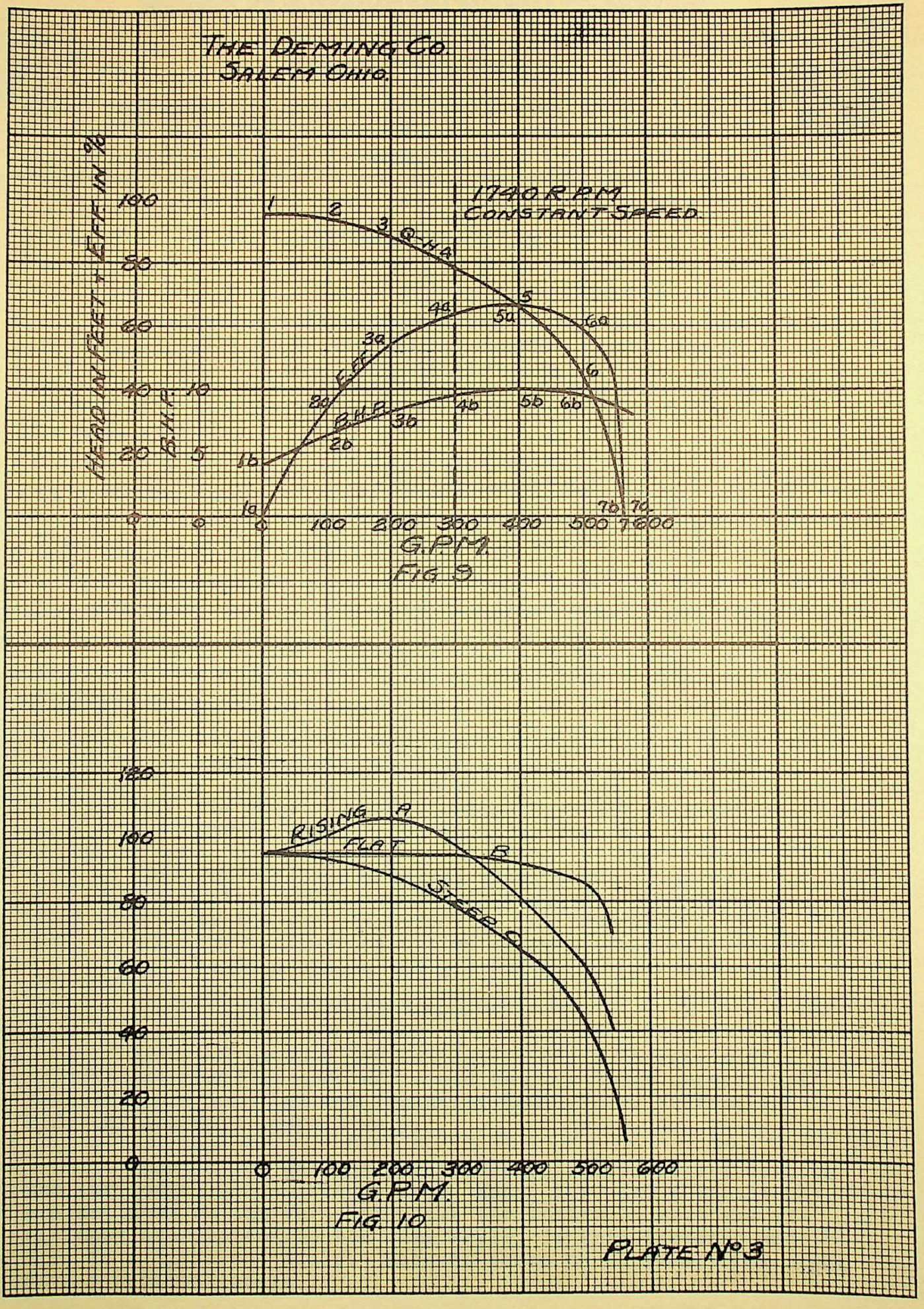

PLATE No 3

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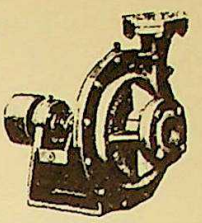

CENTRIFUGAL PUMPS - THEIR THEORY AND APPLICATION

Letter No. 4

Performance Curves

The performance of nearly any machine can be most clearly illustrated by the use of diagrams consisting of curves or straight lines, or combinations of both. These curves are called Characteristics -- they are indeed characteristic of the machine to which they apply, and their name is well chosen.

After a number of tests have been made of the machine in question, the characteristics are constructed by plotting on cross-sectional paper the various quantities involved, and drawing a curve through the several points thus obtained.

Referring to centrifugal pumps, there are three characteristic curves that give all the necessary information.

A pump is usually tested at a constant speed, and therefore these three curves are on that basis. They are as follows:

(1) The Quantity-Head (Q-H) curve, showing the relation between Quantity (or Capacity) and Head.

(2) The Power curve, showing the relation between Power consumed and Quantity pumped.

(3) The Efficiency curve, showing the relation between Efficiency and Quantity. This illustrates the efficiency of the pump when pumping the various quantities against the corresponding heads shown on the Q-H curve.

---o---

Figure 9 shows a typical set of characteristics of a centrifugal pump, at constant speed, and the three curves just described are labelled Q-H (Quantity-Head), H.P. (Power) and Effy (Efficiency) respectively.

Quantity or gallons per minute (G.P.M.) are set down on the horizontal scale (abscissae). Head, H.P. and % Efficiency are set down on the vertical scale (ordinates). Head and efficiency are plotted to the same scale, in this instance, and H.P. to a different scale. Each of these three quantities might have a separate scale.

The Q-H curve is plotted between G.P.M. and Total Head in feet.

The Power curve is plotted between G.P.M. and H.P. (Horse Power consumed by the pump.)

The Efficiency curve is plotted between G.P.M. and percent efficiency.

The curves are usually plotted on the basis of constant speed, and from these curves others can be calculated for different speeds.

Starting then, with a pump test, let us suppose readings have been taken and results calculated for seven so called points and the following data obtained:

Point	G.P.M.	Total Head in Ft.	% Efficiency	B.H.P.	R.P.M.
1	0	95	0	4.2	1740
2	100	94	37	6.4	1740
3	200	88	55	8.1	1740
4	300	78	64	9.25	1740
5	400	66	66	10.1	1740
6	500	44	60	9.3	1740
7	560	10	16	9.0	1740

We will first describe the construction of the Quantity-Head or Q-H curve.

---o---

Q-H CURVE

On a sheet of cross-section paper Fig. 9, mark off a horizontal scale representing Gallons per Minute - In our example, every half inch represents 100 G.P.M. Mark off also a vertical scale representing Total Head in Feet.

Starting at zero G.P.M., count the graduations vertically, and make a mark on the paper representing 95 ft. which is the head produced at shut-off, namely with the discharge valve of the pump entirely closed. This is point 1.

Next, set down point 2, or 94 ft. over 100 G.P.M.

Next, locate point 3 or 88 ft. over 200 G.P.M.

In like fashion, points 4 to 7 inclusive are determined, and these points should all be numbered for reference, so as to avoid confusion. Draw a smooth curve passing through these seven points. This is the Q-H curve.

EFFICIENCY CURVE

The next step is to lay off a verticle scale representing Percent Efficiency. In our example, the same scale is used as for Feet Total Head. At point 1, zero G.P.M. are discharged, and therefore the pump produces no useful work. Therefore, the efficiency for G.P.M. is zero, and the Effiicency Curve starts at zero, which locates point 1a.

The efficiency when pumping 100 G.P.M. is 37%. Lay off 37% over 100 G.P.M. This gives point 2a. Locate point 3a, or 55% over 200 G.P.M. etc. In like manner points 4a to 7a inclusive are located, and a smooth curve drawn through them gives the desired Efficiency Curve of the pump.

---o---

POWER CURVE

Step off a third vertical scale representing B.H.P. (Brake Horse Power). Locate point 1b. or 4.2 B.H.P. over zero G.P.M. Locate point 2b, or 6.4 B.H.P. over 100 G.P.M. and point 3b, or 8.1 B.H.P over 200 G.P.M. Similarly locate points 4b to 7b. Then draw a smooth curve through all these points and the desired Power Curve is obtained.

---o---

DISCUSSION OF THE CHARACTERISTICS

The Q-H Curve

From the tabulated data on page 2 of this letter, it is, of course, possible to read certain quantities direct. For instance, at 1740 R.P.M., the pump will deliver 400 G.P.M. against 66 ft. Head. From the Q-H curve, however, it is possible to determine any number of intermediate heads and capacities, which would be impossible with a table. For instance, draw a line vertically from any desired G.P.M. to intersect the Q-H curve, and a line horizontally from that intersection to the left where it touches the Head Scale determines the Total Head against which the pump will elevate the G.P.M. in question if operated at 1740 R.P.M.

For example, the head corresponding to 350 G.P.M. is not given in the table. From the characteristics, however, by projecting vertically upward from 350 G.P.M. to the Q-H curve, and reading across horizontally to the left, we find the head is 72.5 ft.

Conversely, projecting horizontally from 60 ft. on the Total Head scale to the Q-H curve, and then downward, we find the corresponding capacity is 435 G.P.M.

In the first case, the pump will deliver 350 G.P.M. against a total head 72.5 ft. and in the second example, if gauges indicate the head is 60 ft. we know the pump is delivering 435 G.P.M.

The several curves composing a set of characteristics can be considered made up of an infinite number of points, touching one another. Therefore, an infinite number of capacities and heads may be read from pump characteristics.

Further inspection of the Q-H curve shows that the capacity of the pump increases as the head decreases, but this is only true within resonable limits. If the head is reduced to zero, a condition never obtained in practice, as some friction is present in all pipe and fittings, the pump could not deliver more than 560 G.P.M.

The highest point in the curve is 95 ft at which no water is delivered. This is the shut-off point, or the maximum head or pressure the pump can produce. From this we see that the valve in the discharge line of a centrifugal pump can be closed entirely without building up a pressure that would damage pump or pipe line. Of course, this would be impossible with a displacement pump, as something would have to break if a valve were closed.

Further inspection shows that 94 ft. is the maximum head against which the pump will deliver an appreciable amount of water (about 100 G.P.M.).

It is almost impossible to ascertain from the Q-H curve what effect will be produced upon the capacity of the pump by an increase or decrease of the head. Suppose the pump is applied for pumping 400 G.P.M. against 66 ft. head. If the head is increased to 72 ft., the capacity will decrease to 355 G.P.M. If the head is decreased to 56 ft., the capacity will increase to about 455 G.P.M.

Thus, capacity decreases with increased head, and increases with decreased head, speed remaining intact.

The rate of change of capacity, with changes in head, depends upon the slope of the Q-H curve.

Centrifugal pumps have three distinct types of Q-H curve, or characteristic. These are best illustustrated in Fig. 10 and are known as follows:

(A) The rising characteristic

(B) The flat characteristic

(C) The steep characteristic

The last is often called a drooping characteristic.

Any one of these three Q-H curves can be obtained by proper pump design -- that is, by selecting certain velocities and vane angles.

The curve we are studying is a good example of a steep characteristic, and the capacity variations are relatively small for a considerable change in head. If the Q-H curve were flat, the changes in capacity would be much greater. For instance, referring to the Flat Characteristic in Fig. 10, it is seen that an increase of a few feet in head would bring about a large drop in capacity.

The Q-H curve in Fig. 9 is a very good curve for a general service pump. If the actual head in the installation turns out to be a little greater, or a little less than that for which the pump was designed, there will be little deviation from the desired capacity. For variable head service, and where minimum variation in capacity is desired, a pump having the steepest possible Q-H curve should be used. The steeper the curve, the less change in capacity there will be for any change in head. The ideal curve for such conditions would be a vertical characteristic, but it is impossible to obtain such a curve in a centrifugal pump.

---o---

THE EFFICIENCY CURVE

The Efficiency curve starts at zero (as no useful work is performed at shut-off), rises to a maximum of 66% and drops as the head decreases, or as the capacity increases. If it were possible to reduce the head exactly to zero, the efficiency would be zero at that point, as no useful work would be performed. At both shut-off, and wide open, or maximum capacity, no useful work is done by the pump, and therefore the efficiency at both these points is zero.

It will be noted that the efficiency curve does not rise to a sharp peak, but that its upper portion is nearly flat. This shows that the pump has good efficiency over a comparatively wide range of capacity and head.

For capacities ranging from 250 to 500 G.P.M., the efficiency is 60% or better, which is a very creditable performance for a commercial pump. This pump should not be applied for quantities less than 250 G.P.M. or greater than 500 G.P.M. Smaller or larger pumps should be used, respectively. In other words, 250 to 500 G.P.M. is the economical range of this pump.

By reading the Q-H and efficiency curves together, as it were, it is possible to readily determine the head for a given capacity, or the capacity for a given head, as well as the corresponding pump efficiency.

---o---

POWER CURVE

Inspection of the H.P. (Horse Power) curve shows that it starts at 4.2 B.H.P. (the power required to rotate the impeller at shut-off) and rises to 10.1 B.H.P., after which it drops off slightly. The power required at 1740 R.P.M., under all conditions of capacity and head is shown by this curve. For instance, reading up from 275 G.P.M. to the H.P. curve, it is found that 9.1 B.H.P. are required. When handling 400 G.P.M. 10 1 B.H P. are required. For greater capacities, or less head, less power is required. This is termed the "Non-overload power characteristic".

In earlier centrifugal pump designs, this type of power curve was not obtained. In fact, in most of them the power curve rose sharply as the quantity increased and as the head was reduced. This was such a bad feature that it placed a large overload on the motor driving the pump, and caused a lot of trouble.

The non-overload power curve permits the use of a smaller motor. For instance, referring to Fig. 42, a 10 H.P. 40° motor can be used, as 10.1 B.H.P. is the maximum power required under any head, speed remaining constant. If the power curve rose to, say, a maximum of 12.5 B.H.P. under low heads, a 15 H.P. motor would have to be used, as the load would be too heavy for a 10 H.P. machine.

---o---

SUMMARY

In summing up, we see that the power curve, taken in conjunction with the Q-H and Efficiency curves, gives a complete record of the performance of the pump. These three curves visualize what the pump will do at constant speed. If one of the four items, namely head, capacity, efficiency and power, is known, the other three can be read directly from these characteristic curves.

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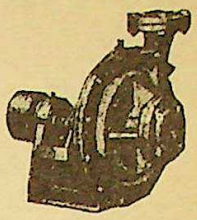


CENTRIFUGAL PUMPS - THEIR THEORY AND APPLICATION

Letter No. 5

The fourth letter was the beginning of the discussion of Performance Curves, and Horse Power and Efficiency were introduced without explanation. In order to make this course complete in itself these terms are now taken up.

Force

Consider a locomotive coupled to a train of cars, and insert a spring balance in the draw-bar. When the train is in motion, the balance will be extended a certain amount. The cause of extension is the force exerted by the locomotive to overcome, broadly speaking, the resistance of the cars due to their weight, and friction in the journals.

This is a good illustration of Force.

Its amount is measured in pounds. Force is the effort that draws the train. Another illustration of Force is the weight of a body as described in Letter Two. It is the intensity with which the body is drawn towards the center of the earth by the attraction of gravitation.

Just as in the case of the draw-bar pull of the locomotive, its amount is measured in pounds.

A mechanical force is any agent tending to produce or change motion. For instance, a Force is also required to stop the motion of the train, as illustrated in braking.

A Force has three characteristics - these are:

(a) Direction
(b) Magnitude, Intensity or Amount
(c) Point of Application

(a) - The first characteristic is the direction in which the body acted upon by force, tends to move.

(b) - The magnitude of the force, as stated above, is measured in pounds.

(c) - The place of application of a force is generally assumed to be a point, such as the center of gravity, or the draw-bar or draft gear, referring to our illustration.

Forces can be conveniently represented by "Straight Lines" and "Arrow Heads". The latter shows the direction of the force, while the length of the line represents the amount or magnitude. (The line is drawn to some convenient scale, wherein the unit length represents so many pounds).

The point of application may be at any point of the line, but is generally assumed to be at one end.

In the accompanying sketch, the force is supposed to act along the line AB in the direction from left to right.

A ——→ B

The length of the line AB represents the magnitude of the force. The point of application may be at A, or B, or any point between. Thus, the line AB fulfills the three characteristics and the force is fully represented.

---o---

The formula, $F = m a$ (Force equals Mass times Acceleration) will be fully explained in the chapter on Centrifugal Force.

Work

Now, suppose the force described above, continues to act, and draws the train a distance of 1000 ft. It performs in so doing what is known as Work.

It is Force acting through Distance

It is, considered from a slightly different point of view, the over-coming of the resistance to the motion of the train over the distance of 1000 ft.

It is the over-coming of the resistance, which is also a force, through a distance.

Mere motion is not work, the body has to overcome distance to perform work.

Work is the product of Force and Distance.

Work is expressed in foot-pounds. If Force is expressed in pounds and distance in feet, it will be seen that their product is naturally foot-pounds.

The unit of work is the foot-pound. Another conception is the amount of work done in lifting a weight of one pound through a vertical distance of one foot, or it is the overcoming of a resistance of one pound through a distance of one foot. The total amount of work is absolutely independent of time.

Power

In the above conception of Work, the element of time has not been taken into consideration. Now if the total work done in foot-pounds in t minutes is divided by this t we have the amount of work done per minute, or the work per unit of time, and this is termed Power.

Power is the rate of performing work. Unit Power is Unit Work done in Unit Time or "One Foot-pound per minute".

$$H.P. = \frac{\text{Total foot-pounds of work per minute}}{33000}$$

The Metric Horse Power is somewhat less, viz 542.47 foot pounds per second, or .9863 English H.P.

Figuring in Electrical Units 746 Watts correspond to the Mechanical Energy of one English H.P. and 736 Watts to one Metric H.P.

---o---

Power Required To Elevate Water

As an example of special interest as we are entering upon a study of pumps, we will investigate the power required to raise water.

Suppose it is required to pump 1000 gallons of water in one minute against a head of 100 ft. that is, 1000 G.P.M. are to be raised 100 ft. high. (We will neglect the resistance encountered in the pipe line to the flow of water, called pipe friction.)

A gallon of cold water weighs approximately 8.33 pounds. Therefore, 1000 gallons weigh 8333 pounds. This weight is to be raised a vertical distance of 100 feet. The resistance to be over-come is the weight of the water, and the distance through which it to be is over-come is the height i.e. 100 ft.

Therefore, the amount of work to be done is

$$8333 \times 100 \text{ or } 833,300 \text{ ft. lbs.}$$

This is the amount of work to be done in one minute. Expressed in H.P. it is

$$\frac{833,300}{33000} = 25.25 \text{ H.P.}$$

This is the work done by the agency used to raise the water, that is the pump. It can be readily seen that it is also the power that would have to be put into the pump in order to enable it to perform this work, if it were a perfect pump, that is, if there were no loses in it. This manifestly, is an impossibility, as losses such as friction are present in all machines.

25.25 H.P. is also the work that the pump delivers, that is, the output, or the power it develops. It is called Theoretical H.P. or more often, Water H.P.

Expressed in a formula

$$W.H.P. = \frac{G.P.M. \times 8\text{-}1/3 \times \text{Head in ft.}}{33000} \qquad (11)$$

By dividing the numerator and denominator by 8-1/3, we have

$$W.H.P. = \frac{G.P.M. \times \text{Head in ft.}}{3960} \qquad (12)$$

or roughly:

$$W.H.P. = \frac{G.P.M. \times \text{Head in ft.}}{4000} \qquad (13)$$

---o---

It has just been pointed out that the Theoretical or Water H.P. is equal to the power applied to the pump, that is, at its shaft, or the driving or operating power, provided there are no friction losses in the pump. However, there are always friction and other losses in any pump.

Calling the externally applied or the driving power the input, it may be conceived that

Input equals Output plus Losses.

It is therefore readily seen that Input is greater than Output. The ratio of output to input is termed the Efficiency, and this is true not only with pumps but for all forms of mechanical machines.

It will be seen that the result is less than unity, or that it is a fraction. Also that, the smaller the losses, the nearer the quotient approaches unity.

$$\text{Efficiency} = \frac{\text{H.P. Output}}{\text{H.P. Output + H.P. Losses}} = \frac{\text{H.P. Output}}{\text{H.P. Input}} \quad (14)$$

Efficiency is always expressed as percent, and a perfect machine, one in which Input = Output would have an efficiency of 100%.

A moment's reconsideration will demonstrate that this example applies to any style pump whether Reciprocating or Centrifugal.

The above formula may be written:

$$\text{H.P. Input} = \frac{\text{H.P. Output}}{\text{Efficiency}} \quad (15)$$

This may also be written:

$$\text{H.P. Input} = \frac{\text{W.H.P.}}{\text{Efficiency}}$$

That is, if the pump in the preceeding example produced an output of 25.25 W.H.P., and its efficiency were 70%, the H.P. required to operate it would be:

$$\text{H.P. Input} = \frac{25.25}{.70} = 36.07$$

H.P. Input to a pump (or other machine) is usually called Brake Horsepower or B.H.P.

Therefore

$$\text{B.H.P.} = \frac{\text{W.H.P.}}{\text{Effy.}} \quad (16)$$

This formula shows, that B.H.P. varies inversely with efficiency. The higher the efficiency, the less power is required to drive. As power is usually an expensive item, this shows the desirability of high efficiency.

The Affinity Law

The Affinity Law, or Law of Proportion, is of great assistance in the many problems that arise in dealing with Centrifugal Pumps. A thorough understanding of this law is one of the greatest importance.

It states the relation between Head, Capacity and Speed or R.P.M. That is, the relation between the first two and speed. After a pump has been tested at a constant speed, and the Q-H Curve plotted, new curves for other speeds can be determined, with out additional tests, by application of the Affinity Law.

Different impellers produce different Q-H Curves.

The fundamental Centrifugal Pump formula which will be taken up in a later letter, teaches us that the head produced is composed of THREE factors: PRESSURE DIFFERENCE, due to the difference of cross-sectional area at the inlet and outlet of impeller passages; CENTRIFUGAL FORCE, due to the speed of the impeller; CONVERSION of the KINETIC ENERGY of the water after it leaves the impeller. If any one of these three changes the Q-H curve will be changed.

The transformation of the kinetic energy, for instance, depends on the diffusor, or casing, and therefore, if even the same impeller is tested in a different casing, or surrounded by a different diffusor, different characteristic curves will be obtained.

In other words, a given impeller will produce results in accordance with the Affinity Law when tested at different speeds but in the same casing only.

While the Affinity Law is a great help to the engineer and designer, it should be used with descretion, since there is a limited range within which it can be applied. Points near the maximum capacity usually do not agree with te Affinity Law. The saying "A little knowledge is a dangerous thing" is particularly true with respect to it. An over zealous salesman for instance can get his company into a lot of trouble by a reckless application of this law.

The Affinity Law states:

(1) The head changes in proportion to the square of the speed.

$$H_1 = H \left[\frac{n_1}{n}\right]^2$$

(2) The capacity changes in direct proportion to the speed.

$$Q_1 = Q \left[\frac{n_1}{n}\right]$$

(3) The Power changes in proportion to the cube of the speed.

$$HP_1 = HP \left[\frac{n_1}{n}\right]^3$$

H, Q and HP are the values found by test at speed n

H_1, Q_1 and HP_1 are the new values when speed is changed to n_1

Application

Let us refer to Fig. 11 and and consider point 5, representing 400 G.P.M. and 66 Ft. Head. If the pump speed is increased 10%, or to 1914 R.P.M., the new head will be

$$x = 66 \left[\frac{1914}{1740}\right]^2 = 80 \text{ Ft.}$$

and the new capacity will be

$$y = 400 \left[\frac{1914}{1740}\right] = 440 \text{ G.P.M.}$$

Locate this new point 5', on the diagram, representing 80 ft. and 440 G.P.M.

Similarly points 3 and 4 increase to points 3' and 4' respectively. A curve is then drawn through points 3' 4' 5', which represents the performance of the pump at 1914 R.P.M.

Roughly speaking, the efficiency for 440 G.P.M. will be the same as for 400 G.P.M., etc.

We can thus draw in the new efficiency curve.

There is a shorter method for securing the characteristics at 1914 R.P.M. Merely transfer point 5 as indicated and through it draw a curve parallel to the curve 1-2-3-4-5 etc. This is close enough for approximations, but not quite so accurate as the curve obtained by transferring several points.

From the new Q-H and efficiency curves the new Power curve can be readily calculated.

---o---

Conversely, if the speed is decreased 10%, or to 1566 R.P.M., applying the Affinity Law, we get an entirely new characteristic 1_1-2_1-3_1 etc.

Point 5, for instance, reduces as follows:

$$\text{New head} = 66 \left[\frac{1566}{1740}\right]^2 = 53.5 \text{ ft.}$$

$$\text{New capacity} = 400 \left[\frac{1566}{1740}\right] = 360 \text{ G.P.M.}$$

The efficiency at 360 G.P.M. will equal the efficiency at 400 R.P.M., etc. From the new Q-H and Efficiency curves, we can calculate the new Power Curve.

---o---

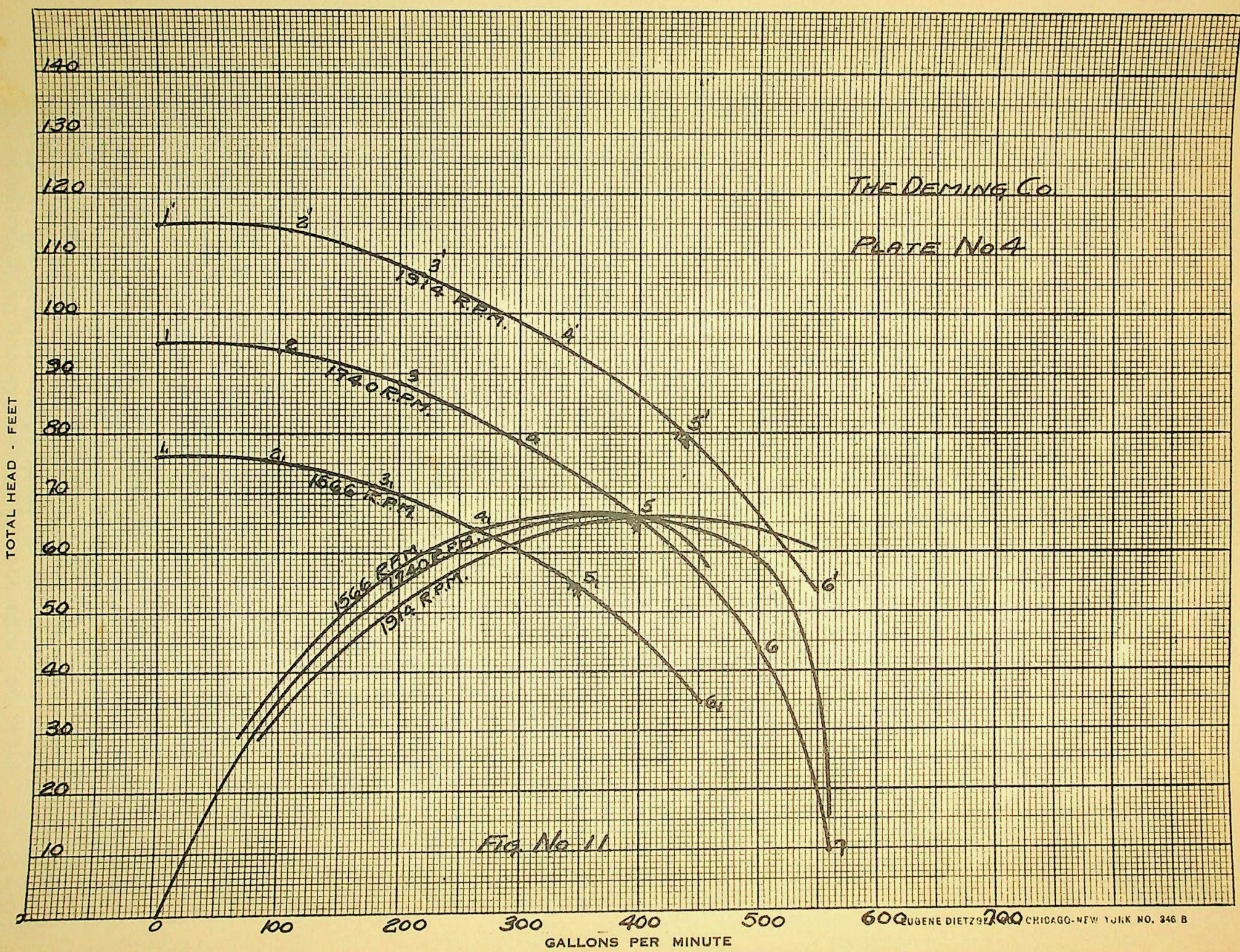
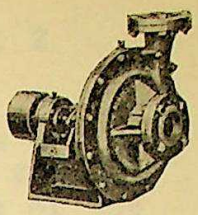

THE DEMING CO.
PLATE No 4
TOTAL HEAD - FEET
GALLONS PER MINUTE
1914 R.P.M.
1740 R.P.M.
1566 R.P.M.
1566 R.P.M.
1740 R.P.M.
1914 R.P.M.
140
130
120
110
100
90
80
70
60
50
40
30
20
10
0
100
200
300
400
500
600
700
EUGENE DIETZGEN CO. CHICAGO-NEW YORK NO. 346 B

Fig. No. 11

THE DEMING COMPANY
(ESTABLISHED 1880.)
SALEM, OHIO, U.S.A.



CENTRIFUGAL PUMPS - THEIR THEORY AND APPLICATION

Letter No. 6

Change in Capacity by Throttling

Referring to Fig. 12, suppose the gate valve in pump discharge pipe is partially closed so that the capacity is only 300 G.P.M., whereas pump with valve wide open had been delivering 400 G.P.M. It will be noted that the pump will deliver 300 G.P.M. against 78 ft. head. That is, 78 ft. is the pressure that will be produced back of the discharge valve, or between the discharge valve and the pump proper.

From Fig. 9 we saw that the power required when pumping 400 G.P.M. was 10.1 B.H.P. The efficiency when pumping 300 G.P.M. against 78 ft. head is 64%, and 9.25 B.H.P. are required. Therefore power can be saved by throttling.

Assume that the pump is installed and working on a system of piping where the head is 66 ft. Its capacity will be 400 G.P.M. at 1740 R.P.M. Now suppose the capacity is throttled to 300 G.P.M. The head indicated by the pressure gauge, which should be located between the pump and the throttle valve, and which previously read 66 ft. will now read 78 ft. This is not a useful head but is literally wasted in overcoming friction due to the restricted area through the valve.

The head against which a pump operates is composed principally of two factors, static and pipe friction. The static head is the difference in elevation between source of pump supply and point of delivery, and the friction is the resistance in the pipe line to the flow of water through it.

In our example, consider that the head is all static such as a pump located alongside of and discharging directly into a standpipe.

It is obvious that the static head is the same, regardless of the G.P.M. pumped, provided the level in the standpipe is maintained constant. The straight horizontal line, marked "66' Static" represents the head of the system or the actual pumping head, and is entirely independent of the pump characteristic.

Referring to Fig. 12, suppose the pump delivers 400 G.P.M. against the existing head, 66 ft. which is all static, with the discharge valve wide open. If it is desired to reduce the capacity to 300 G.P.M., the discharge valve is partially closed until the capacity is reduced to the desired amount But the pressure gauge mounted at the pump discharge shows 78 ft. whereas obviously the the actual head in the system remains the same (66 feet).

The difference, or 12 ft., is wasted in the throttle valve or expressed differently, is the head absorbed in overcoming the resistance to the flow of water due to the partial closing of the valve. The efficiency when discharging 300 G.P.M. is shown to be 64%, from the Efficiency Curve, but this corresponds to 78 ft. Head. The B.H.P. therefore is

$$\frac{300 \times 78}{3960 \times .64} = 9.25$$

The actual, useful head overcome, however, is only 66 ft. Therefore the useful work done by the pump is only

$$\frac{300 \times 66}{3960} = 5.02 \text{ H.P.}$$

But the power consumed is 9.25 B.H.P.

Therefore the useful efficiency is not 64%,

$$\text{but} \quad \frac{5.02}{9.25} = 54.3\%$$

This may be termed the plant efficiency, and it is very important, when estimating upon the efficiencies secured by throttling a pump, to clearly specify whether reference is made to the "efficiency when discharging against the increased pressure due to throttling" or to the "plant efficiency".

---o---

Let us now take a different case, namely, consider that the original head in the piping system, 66 ft., is composed as follows:

Static 40 ft.

Friction 26 ft.

When only 300 G.P.M. are passing through the pipe line, the friction head will be only 14.7 ft. and the total head 54.7 ft.

---o---

The frictional resistance to the flow of water through a pipe varies as the squares of the velocities or as the squares of the respective capacities (to be more fully explained in a later letter).

That is, $26 : x = 400^2 : 300^2$

$$x = \left|\frac{300}{400}\right|^2 26 = 14.7$$

In Fig. 12 draw a horizontal line, 40 ft. Static, representing static head. Calculate the friction for 100, 200, 300, 400 G.P.M. etc. and lay off these values above the line of 40 ft. static head. A curve drawn through these points shows the head in the piping system for any capacity, and is therefore the "Head of System" curve. It has nothing whatever to do with the pump characteristics, as it is entirely independent of them.

For instance, reading up from 325 G.P.M. to the Head of System curve, and projecting across to the Head Scale, we find the total head to be 57.5 ft.

---o---

Returning to throttling the pump, the head produced by the pump is still 78 ft. and the power consumed 9.25 B.H.P. But the head in the system corresponding to 300 G.P.M., is from the Head of System curve, only 54.7 ft. Therefore the useful work done by the pump is 4.14 H.P., and the efficiency is

$$\frac{4.14}{9.25} = 44.7\%$$

In this case the head wasted in throttling is 78 ft. minus 54.7 ft. or 23.3 ft.

---o---

From these two examples it is evident that when capacity is to be reduced by throttling, the most efficient results are obtained when the head is all static or nearly so. Under these conditions only is the least head wasted in the throttle valve.

On the other hand, the greater the percentage of friction in the total head, the greater is the head wasted in the throttle valve, and the lower is the useful efficiency of the pump.

---o---

In most installations throttling is the only method of reducing capacity. There are two other methods of decreasing capacity. First, by reducing the R.P.M. Second, by decreasing the impeller diameter.

DECREASING CAPACITY BY SPEED REDUCTION

Case 1, Head all Static

Let us suppose it is desired to decrease the capacity to 300 G.P.M., the head, of course, remaining 66 ft. as there is no friction. Locate a point in Fig. 12 representing 300 G.P.M. 66 ft. and mark it by a little cross-hatched corner.

We saw from Fig. 11 and the discussion relating to it, that a change in R.P.M. produced a new Q-H curve parallel to the existing curve. Thus curves 1'-2'-3'-4'-5', etc. and 1_1-2_1-3_1-4_1-5_1-6_1 parallel curve 1-2-3-4-5-6.

Returning to Fig. 12 it is evident that some lower speed will correspond to a new Q-H curve, parallel to curve A, and passing through the desired point, viz. 300 G.P.M. 66 ft. In other words, at this speed, which we are now going to determine, the pump will deliver 300 G.P.M. against 66 ft. head.

The Affinity Law is resorted to, and several trials are made. For instance, we see from Fig. 11 that at 1566 R.P.M. the pump delivers 300 G.P.M. against only 60.5 ft. head. It is evident, therefore, that the desired speed lies somewhere between 1740 and 1566 R.P.M.

Now draw a pencil Q-H curve through our desired point, and parallel to curve A, - call this curve B.

Let us try a speed of 1620 R.P.M. Then, by the Affinity Law, 300 G.P.M.-78 ft. on Curve A reduces to 279 G.P.M. - 67.5 ft. Locating this point it is found that it lies practically upon curve B; it is only a trifle below it, in fact, showing that our trial speed is a trifle low. Let us call the correct speed 1625 R.P.M., which is close enough for all practical purposes.

Therefore, if the pump speed is slowed down to 1625 R.P.M., the pump will deliver 300 G.P.M. against 66 ft. head and no more. The throttle valve is, of course, left wide open, and as there is no throttling no head is wasted.

By applying the Affinity Law again, we find that 300 G.P.M. - 66 ft. on B corresponds to 321 G.P.M. - 75.5 ft. on A, which in turn corresponds to approximately 65% efficiency. The efficiency, therefore, for 300 G.P.M. - 66 ft. is 65%, which is the useful efficiency, as against 54.3% which was the useful efficiency when throttled.

The comparison is even more striking when considering the power required:

9.25 B.H.P. was required when throttled

7.70 " is required at the lower speed

1.55 " is saved.

This shows very clearly that more power can be saved by speed reduction than by throttling. The difference is even greater where the head is part static and part friction.

---o---

Case 2 - Head Part Static and Part Friction

Let us assume that it is desired to decrease the capacity to 300 G.P.M. by decreasing the speed. Referring to Fig. 12, from the Head of System curve we find the total head corresponding to 300 G.P.M. is 54.7 ft. We must now determine a lower (than 1740 R.P.M.) speed that will produce a new Q-H curve through the point 300 G.P.M. - 54.7 ft.

From Figures 12 and 11 it will be apparent that this desired speed is lower than 1625 R.P.M. and 1566 R.P.M.

Draw a pencil Q-H curve through the desired point, and call it curve "C".

After several trials, it is found that 1500 R.P.M. produces the desired Q-H curve, that is, at 1500 R.P.M. the pump will deliver 300 G.P.M. against 54.7 ft. head. Transferred back to 1740 R.P.M.,this corresponds to 348 G.P.M. - 73 ft., and 65.5% efficiency.

Therefore, when running at 1500 R.P.M. and delivering 300 G.P.M. against 54.7 ft. head, the efficiency will be 65.5% and 6.32 B.H.P. will be required.

Contrast these results with 44.7% effieiency, and 9.25 B.H.P. respectively, which were obtained when the pump was throttled. Nearly 3 B.H.P. (2.93 B.H.P.) is saved by speed reduction.

---o---

From these examples it is seen that when it is desired to reduce the capacity of a pump, more efficient results can be obtained by speed reduction than by throttling The efficiencies cited refer, of course, to the pump only. If the changes are secured by using armature resistance with a (direct) current motor having a normal speed of 1740 R.P.M., the gain in efficiency is somewhat off-set by the decreased efficiency of the motor, but if a motor having a normal speed of 1100 R.P.M. is used, and the several speeds we have mentioned are obtained by field control, the motor efficiency is not so greatly impaired, and the gain in efficiency applies approximately to the over-all efficiency also.

INCREASE OF CAPACITY BY INCREASE IN SPEED

Case 1 -- Head all Static

Referring to Fig. 12 it is desired to obtain 450 G.P.M. against 66 ft. head. Locate this point on Fig. 12 and draw through it another pencil Q-H curve parallel to curve A, call it Curve D. Evidently, the pump at some speed higher than 1740 R.P.M. will produce a Q-H Curve approximating Curve D. By several trials we find the desired speed is 1820 R.P.M. and the efficiency about 65%. This will enable us to calculate the B.H.P. required.

---o---

Case 2 -- Head part Static and part Friction

From Fig. 12 the total head in the system for 450 G.P.M. is found to be 73.2 ft. Proceeding in similar fashion, we find that a speed of 1875 R.P.M. will give us a new Q-H curve passing through the point 450 G.P.M. - 73.2 ft. and that the efficiency will be approximately 65%. This is curve E.

It is thus seen that a greater increase in speed is necessary to bring about a given increase in capacity, where there is friction as compared with a head 100% static.

---o---

EFFECT OF CHANGE OF HEAD WHERE PIPE FRICTION IS PRESENT

We saw in Letter No. 4, Paragraph 1, Page 4, what effects were produced by changes of head, but considered the head all static, that is, we neglected friction. This element, however, has a marked effect in controlling pump capacity under varying heads as we will now demonstrate. Referring to Fig. 13, we have reproduced the same Q-H curve at 1740 R.P.M. that is illustrated in Fig. 11 and Fig. 12. Let us consider the same point, 400 G.P.M. - 66 ft.

It is assumed that the pump is delivering to a stand-pipe through a pipe line of such length and size that the friction is 26 ft. The water level is maintained at such a point that the static head is 40 ft. The total head therefore, is 66 ft.

Let us assume that it is desired, for example, to carry a 20 ft. lower level in the stand-pipe, thus making the static head only 20 ft. The pipe line is just the same, therefore the friction head, for 400 G.P.M., would be 20 + 26 = 46 ft. From this it would seem, by inspecting the Q-H curve that the capacity would increase to about 495 G.P.M. However, for greater capacity (than 400 G.P.M.) the friction, and consequently the total head would be greater and thus the actual increase in capacity will be less than 95 G.P.M.

In Fig. 13, plot the Head of System Curve, just as previously described, except lay off the various friction heads above the horizontal line, 20 ft. Static, as a base. We thus obtain the new Head of System Curve, for 20 ft. Static Head. Where this intersects the Q-H Curve of the pump determines the G.P.M. that will be delivered; in this case, about 462 G.P.M.

We know from the Head of System Curve that the total head for 462 G.P.M. is 56 ft. and we also know that the pump will deliver 462 G.P.M. against 56 ft. head.

By plotting the Head of System Curve we are saved the tedious process of making several guesses as to what the new capacity will be and then figuring total heads and seeing whether they lie in the Q-H Curve. The method just illustrated is direct and concise.

The point in question must lie on both the Head of System and the Q-H Curve. In order to fulfill this condition it must lie at <u>their intersection.</u>

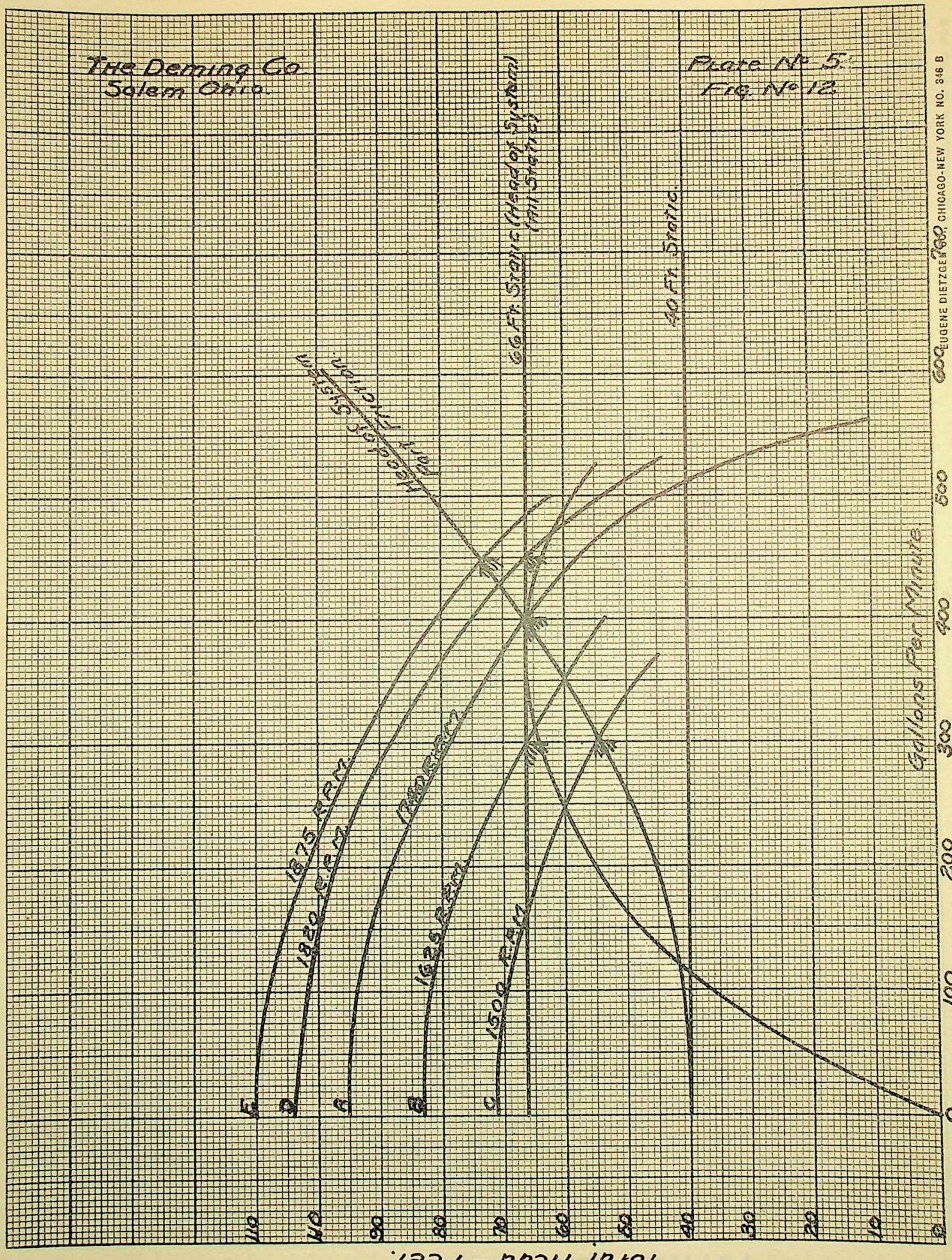
---o---

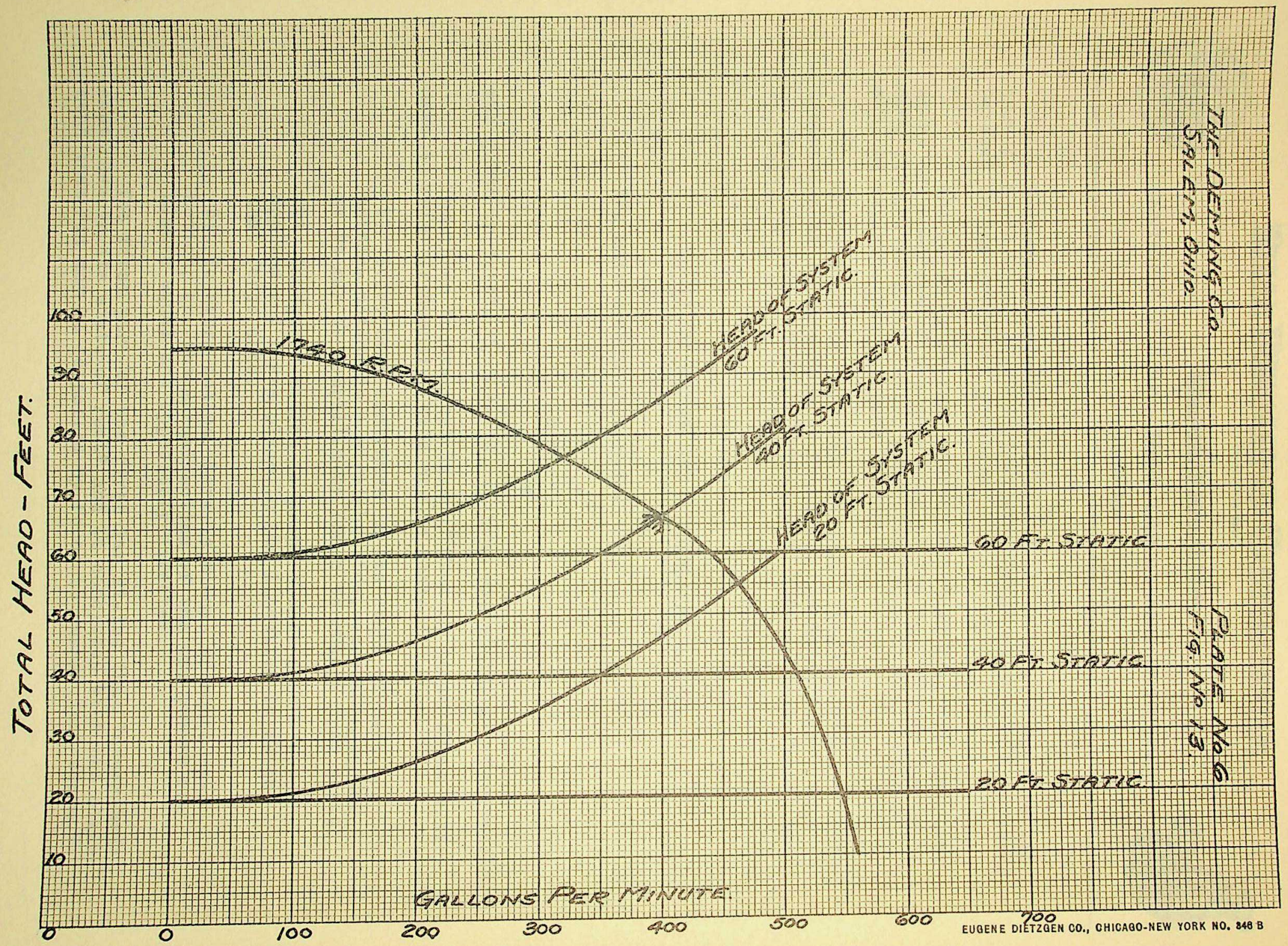
As another example, suppose the level in the stand-pipe is to be maintained 20 ft. <u>higher</u>. The pipe line, and therefore the friction, remains the same. The total head, then, for 400 G.P.M. will be 60 + 26 = 86 ft., and from the Q-H Curve it would appear that the pump would deliver approximately 225 G.P.M. However, the actual capacity will be greater than this, and may be determined as follows:

In Fig. 13 draw the horizontal line, 60 ft. Static, and lay off the "Head of System - 60 ft. Static" Curve about it as a base. Its intersection with the Q-H Curve determines the new capacity, namely approximately 315 G.P.M.

It is thus seen that pipe friction exercises a balancing effect when there are changes in head. When the static head is reduced, the resulting capacity is <u>less</u>, and when the static head is increased, the resulting capacity is <u>greater</u> than might otherwise be expected.

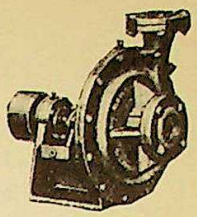
By referring to Fig. 11 the efficiencies for 462 G.P.M. and 315 G.P.M. can be read, and, knowing the corresponding heads the B.H.P. can be calculated.


The Deming Co.
Salem Ohio.
Plate No 5.
Fig No 12.
66 Ft. Static (Head of System)
(All Sizes)
40 Ft. Static.
Heads of Sizes Given Each Curve.
1675 R.P.M.
1820 R.P.M.
1625 R.P.M.
1500 R.P.M.
Impeller
E
D
A
B
C
Total Head - Feet.
Gallons Per Minute
0
100
200
300
400
500
600
700
10
20
30
40
50
60
70
80
90
100
110



EUGENE DIETZGEN CO., CHICAGO-NEW YORK NO. 848 B

THE DEMING COMPANY
ESTABLISHED 1880
SALEM, OHIO, U.S.A.



CENTRIFUGAL PUMPS - THEIR THEORY AND APPLICATION

Letter No. 7

Performance Curves have been dealt with at length in letters 4, 5 and 6 in order to give information which could actually be used in the every day problems.

The grouping of such curves, obtained from various diameters of impellers of the same pump at the most common speeds -- 860, 1150, 1750 R.P.M. etc., generally known as "Data Sheets" will be explained in a future letter.

The subject of this and the next letter will be a further discussion of fundamentals of Mechanics and Hydraulics. This is necessary in order to make "Centrifugal Pumps, Their Theory and Application" a course in itself.

---o---

FALLING BODIES AND ACCELERATED MOTION

The main hydraulic formula

$$h = \frac{v^2}{2g} \qquad (3)$$

was given in letter 2, Sheet 4, without explaining its derivation.

The three fundamental laws of Newton, to which we will have to refer frequently, will now be stated:

(1) Every body continues in its state of rest, or uniform motion in a straight line, except in so far as it may be compelled by force to change that state.

(2) Change of motion is proportional to the force applied and takes place in the direction in which that force acts.

(3) To every action there is always an equal reaction; or, the mutual action of the two bodies are always equal and oppositely directed.

The first part of the first law states that "every body continues in its state of rest" and it concludes with the statement that "it may be compelled by force to change that state."

In accordance with this law, any body upon the earth's surface is held there and an effort is required to lift or move it; it is held there by the attraction of gravity, which, as explained before, is strongest at the earth's surface.

Again, the first statement of Newton says that, "Every body will continue in a uniform motion unless compelled by force to change that state." Should therefore a force act upon a body, free to move, then it will move forever with the same velocity unless "acted upon by another force."

Supposing such a force did act upon a body and at the end of one second, the same force were to act again, then the velocity at the end of the second second would be twice as great as at the end of the first second. Should the same force act again it is obvious that the velocity at the end of the third second would be three times as much as at the end of the first second.

Thus, if a constant force acts upon a body, free to move, the velocity of the body at any time will be "the product of the velocity at the end of the first second and the time in seconds."

Such a constant force is called a "Constant Accelerating Force"; the force could also be a "Constant Retarding Force", if the velocity were constantly decreased instead of increased.

Velocity at any time = Velocity at end of first second times Time in seconds.

The unit of time in mechanics is the "second" and the unit of distance in the English Standard of measurement is "feet"; we, therefore, have "feet and second" Units.

Acceleration is the rate at which the velocity of a body increases in unit time; thus, unit acceleration is an increase in velocity of one foot per second per one second.

1 ft. per sec. per sec. is also written 1 ft./sec.^{2}

Now if the acceleration in unit time is more than 1 ft., for instance, 6 ft. per sec., we would have at the end of three seconds the velocity

$$v = 6 \times 3 = 18 \text{ ft./sec.}$$

and the mistake should not be made of writing

$$v = 6 \times 3^2$$

which would give a wrong value.
It must be remembered that it is 6 ft. per one second times the number of seconds.

THE FORCE OF GRAVITY

We are now ready to explain the force of gravity, denoted by the letter g.

When a body is dropped from a height, the velocity of this body constantly increases as it approaches the ground, since the attraction of the earth, or force of gravity is constant (at that point) and acts as a constant accelerating force upon the body.

It has been demonstrated that all bodies, no matter what their weight or shape fall equally fast in a vacuum, and generally speaking, in the air also.

The apparent discrepancy of a relatively heavy body, such as a stone falling faster than a light body, such as a leaf, is explained by the air resistance encountered.

An iron ball and a wooden ball of the same dimensions, when dropped from an equal height, will strike the ground at the same time, but while they strike at the same time, the "impact", with which they strike is quite different, as we all know from similar experience in life.

We will apply this right now to the centrifugal pump.

Let the wooden ball represent a volume of water and the iron ball an equal volume of heavier liquid, say sulphuric acid.

They would both reach the earth with the same velocity and by reversing the process, they would have to be thrown up into the air with the same starting velocity to reach the same height.

Note well, the iron ball does not start faster than the wooden ball - therefore: "In order to reach the same height, water and sulphuric acid must leave the impeller at the same speed".

In other words, the impeller referred to in Letter No. 2, Sheet No. 3, with a circumferential velocity of 91.5 ft./ sec. will operate against a head of

$$h = \frac{v^2}{2g} = \frac{91.5^2}{2 \times 32.16} = 130 \text{ ft.},$$

no matter if water, sulphuric acid, oil, mercury, or any other liquid is being handled, disregarding, of course, viscosity (stickiness) which has a decided influence.

It is different, however, so far as impact is concerned. Therefore, we must use more force in throwing the iron ball than the wooden ball.

Applying this reasoning to the two liquids, means that it will take more power to rotate the impeller at the same speed when pumping sulphuric acid than when pumping water.

While this is very simple, mistakes are often made in this respect.

---o---

Careful experiments have shown that the force of gravity g on a freely falling body is the equivalent of giving a velocity of 32.16 ft. in one second to the body.

Under "Weight and Mass" it was stated that g varies, having different values on, above and below the earth's surface.

We should now add that it also is different at the Equator than at the North or South Poles; it varies with the Latitude.

It is at the Equator 32.0902 and at the Poles 32.2549, but in ordinary mechanical problems involving the force of gravity, the value g equals 32.16 is used.

Taking the case of a body starting from a position of rest and falling freely, the velocities obtained at different times, according to the foregoing and proven by tests are as follows:

Time	Vel. in feet per second.
At end of 1st second	32.16 ft.
" " " 2nd "	32.16 + 32.16 = 64.32 ft.
" " " 3rd "	32.16 + 32.16 + 32.16 = 96.48 ft.

This we can formulate

$$v = 32.16 \times t$$

or

$$v = g\,t \qquad (17)$$

where v represents the velocity at the end of the first second and t the time seconds.

The space through which the body falls during the first second is naturally equal to

$$\frac{g}{2}$$

since it starts with zero velocity and must be the average between this and the final velocity since the acceleration is constant.

This tabulated again similar to the final velocities, we have

Time	Space fallen through
At end of 1st sec.	$\frac{0 + 32.16}{2} = 16.08$ ft.
" " " 2nd "	$16.08 + \frac{32.16 + 64.32}{2} = 64.32$ ft.
" " " 3rd "	$64.32 + \frac{64.32 + 96.48}{2} = 144.72$ ft.
etc.	

It is:

$\frac{64.32}{16.08} = 4 = 2^2$ 2nd sec.

$\frac{144.72}{16.08} = 9 = 3^2$ 3rd sec.

$\frac{257.28}{16.08} = 16 = 4^2$ 4th sec.

2^2, 3^2 and 4^2 are the squares of the number of seconds during which the body fell.

It is obvious from this that a body free to move, will in a given time fall a distance to 16.08 (or $\frac{g}{2}$) multiplied by the squares of the time in seconds.

We can formulate this:

$$s = \frac{g}{2} t^2 \qquad (18)$$

where s represents the distance fallen through in t seconds.

Transposing formula (18) we get:

$$t^2 = \frac{2s}{g} \quad \text{or} \quad t = \sqrt{\frac{2s}{g}} \qquad (19)$$

By inserting the value of t in equation (17) we get:

$$v = g \times \sqrt{\frac{2s}{g}}$$

Squaring the right and left side of the equation gives:

$$v^2 = g^2 \times \frac{2s}{g} = \frac{g^2\, 2s}{g}$$

$$v^2 = 2gs \qquad (20)$$

In hydraulics, s the distance fallen through in feet is universally called "Head" and designated by the letter H or h.

Therefore, equation (20) readily becomes

$$v^2 = 2gh$$

$$\text{or } v = \sqrt{2gh} \qquad (4)$$

which is the equation set down in Letter 2, Sheet 4.

By extracting the square root of the constant factor 2 g we can rewrite equation (4)

$$v = 8.02 \sqrt{h}$$

This is a very convenient form for slide rule use.

While the study of the uniformly retarded motion is very interesting in itself, this motion represents the reverse of uniformly accelerated motion, and its formulae are not encountered in the further study of Centrifugal Pumps.

--o--

ENERGY

From a mechanical point of view, a body possesses Energy when it has the capacity or possibility of performing work, that is, being able to overcome resistance through distance.

Generally speaking, energy is something that is imparted to a body by doing work upon it, for instance, when a weight is raised or put into rapid motion, or when a spring is compressed. The energy is given out when the weight falls, or is brought to a stop, or when the spring uncoils, thus doing work.

The meaning of energy and work are the same, but it must be realized that energy is not work; energy is only the capacity of doing work, while work represents something already accomplished; transformation of energy into work is only possible by over-coming resistance. A freely falling body is stored up with energy but it does not do any work until it meets resistance, for instance striking the earth. Work cannot be done without motion.

There are two kinds of mechanical energy: Potential (Static) and Kinetic (Dynamic).

---o---

Potential energy is latent energy; it is the power of performing work, the energy possessed by a body due to its position or condition.

A body possesses potential energy when it is so located that a force acting upon it will produce motion as soon as the restraining force or object has been removed.

The following will illustrate this clearly:

A ball weighing 10 pounds is suspended from a cord so that it hangs 30 ft. in the air above the ground. As soon as the cord is cut, the ball will fall and overcome the air resistance through a distance of 30 ft. The amount of work performed depends on the weight of the ball and the distance traveled by it during the fall. It is 10 lbs. x 30 ft. or 300 foot-pounds. That is:

$$E_{potential} = E_p = w\,s \qquad (22)$$

where w = Weight of the object in pounds.

s = Distance traversed in feet.

In order to raise the ball to a height of 30 ft. an expenditure of 10 lbs. x 30 ft. = 300 foot-pounds is necessary and when the ball falls, just this amount of energy which has been stored within, is released. That is, hanging on the cord, the ball has the capacity of doing 300 foot-pounds of work. The potential energy of the ball is therefore 300 foot-pounds. The instant the cord was cut, potential energy was liberated and motion was made possible thus generating kinetic energy and enabling work to be done. The potential energy was entirely changed into kinetic energy as soon as the ball struck the ground.

KINETIC ENERGY

In order to understand kinetic energy and the derivation of the formula for Ek we shall have to consider what is known as the "Conservation of Energy".

(Experience has demonstrated the fact) science tells us that energy cannot be created or destroyed. It can be changed, or transformed, or spent, but it cannot be made to disappear. It may not be apparent but it has merely changed its form. Energy simply exists, and is present everywhere, like matter. It can never be destroyed.

Ek is the power of doing work that is possessed by a body in motion. A moving ball, automobile, train, flywheel, current of air, jet of water, the water leaving the impeller of a centrifgal pump, all have kinetic energy.

The formula for kinetic energy is easily derived by starting out with the formula for potential energy.

$$Ep = w\ s \tag{22}$$

For distance s we substitute the height h which is also a distance.

From formula (3) letter 2, we know that

$$h = \frac{v^2}{2\ g} \tag{3}$$

For w in formula (22) we substitute

$$w = m\ g \tag{5}$$

and obtain

$$E = m\ g\ x\ \frac{v^2}{2\ g} \quad \text{or}$$

$$E = \frac{m\ v^2}{2}$$

The v indicates that motion prevails and instead of potential energy, we have its transformation kinetic energy

$$Ek = \frac{m\ v^2}{2} \tag{23}$$

MOMENTUM

Momentum, or quantity of motion, is the product of the mass of a body and its velocity at any instant

$$M = m\,v = \frac{w}{g}\,v \qquad (24)$$

Since the moving force equals mass by acceleration, $F = m\,a$, and if the velocity acquired in t seconds = v, or $a = \frac{v}{t}$ then

$$F = \frac{m\,v}{t} \text{ or } Ft = m\,v$$

That is, the product of a constant force and the time in which it acts equals numerically the momentum.

If $t = 1$ second, we can write $M = m\,v$ (24)

From this consideration, momentum might be defined as numerically equivalent to the number of pounds of force that will stop in one second a body moving with velocity v, or the number of pounds of force which actually, during one second, will give the body the velocity v

---o---

MOMENT OF FORCE

The moment of a force with respect to a point is the product of the force multiplied by the perpendicular distance from the given point to the direction of the force. In the illustration (Fig. 14, Plate #7) the moment of the force F with relation to point A is F x AB.

The perpendicular distance AB between the fulcrum A and the Force F is called the lever-arm (l) of the force. If instead F, a force F' acts upon B in the direction indicated, the lever arm would be l'. The moment is the measure of the tendency of the force to produce rotation about the given point, which is termed the center of moments. If the force is measured in pounds and the distance in feet, the moment is expressed in foot-pounds.

$$Mf = F\,l \text{ foot-pounds} \qquad (25)$$

The moment of the resultant of any number of forces acting together in the same plane is equal to the algebraic sum of the moments of the separate forces.

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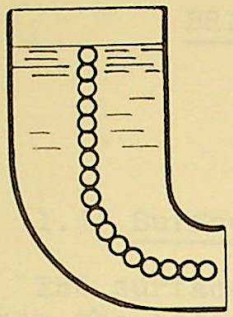

FIG. 20

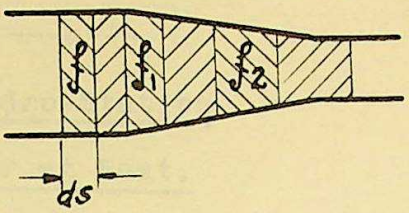

FIG. 21

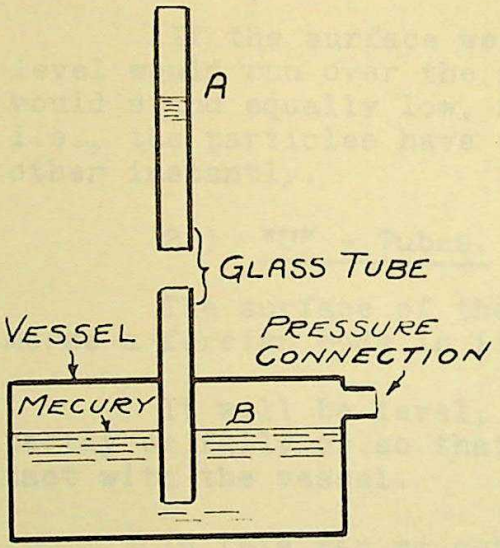

FIG. 22

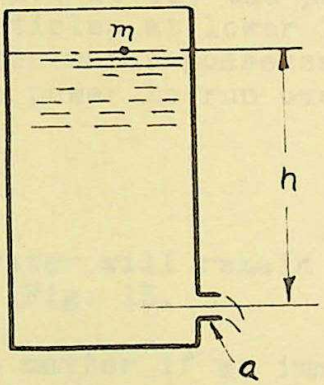

FIG. 23

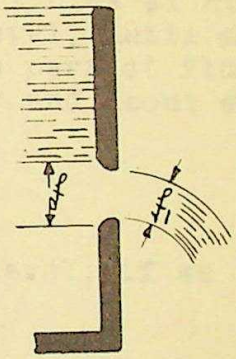

FIG. 24

FIG. 25

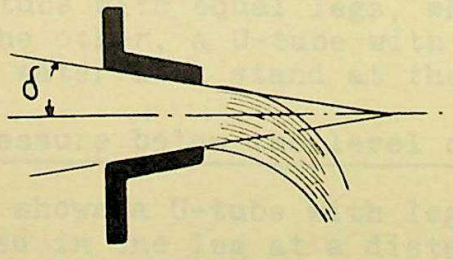

FIG. 26

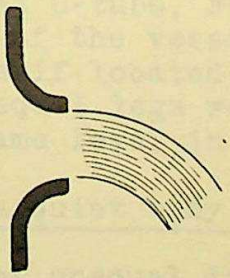

FIG. 27

PLATE No. 8

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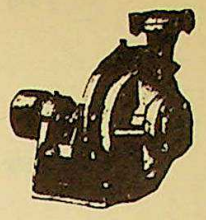
CENTRIFUGAL PUMPS - THEIR THEORY AND APPLICATION
Letter No. 8



BRIEF DISCUSSION OF HYDRAULICS

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Hydro-Statics

1.) Surface of Water at Rest.

The surface of a volume of water contained in a vessel and under the sole influence of gravity is, when at rest, horizontal, or level.

If the surface were not level, the particles at higher level would run over the particles at lower level, until all would stand equally low, since water possesses perfect fluidity, i.e., the particles have the power to run over and about each other instantly.

2.) "U" - Tubes.

The surface of the water will remain level, if we immerse a foreign body in it. Fig. 15.

It will be level, no matter if we immerse this body only partly or fully or so that its sides come in part or full contact with the vessel.

In this way we obtain the so-called U-tube, Fig. 16. A symmetrical body immersed in the middle of the vessel would illustrate a U-tube with equal legs, while if located closer to one side than the other, a U-tube with unequal legs would result; still the water will stand at the same level in both legs.

3.) Pressure below the level of a quiet body of water.

Fig. 17 shows a U-tube with legs of unequal diameter and a piston inserted in one leg at a distance h from the top or surface of the water.

We assume that this piston is weightless and of the infinitely small thickness dh. It is also to be in frictionless contact with the tube wall and consequently the equilibrium is not affected.

We now remove the volume of water standing above the piston and replace it with a force P of suitable magnitude, acting vertically upon the piston so as not to disturb the equilibrium.

The quantity of water just removed, simply pressed upon the piston because the sidewalls of the U-tube did not offer it any support.

This proved that the force P must be equal to the weight of the water, the place of which it takes.

If λ represents the heaviness or the weight of unit volume of water, and f represents the area of the piston we have

$$P = f h \lambda \qquad (26)$$

Upon unit area at the depth h below the surface, we have in a vertical direction the unit pressure

$$P = h \lambda$$

---o---

Fig. 18 shows a similar illustration to the previous one but going a step further. The middle leg is filled with water to a height h and the two outside legs are equipped with pistons, weightless and frictionless as before. These pistons are on the same level, and forces P_1 & P_2 are applied to balance the water column h.

If f_1 & f_2 represent the areas of the pistons, we have as per equation (26) the following relations:

$$P_1 = f_1 h \lambda$$

$$P_2 = f_2 h \lambda$$

If we remove the forces P_1 & P_2 and fill the outside legs with water up to the height h, the equilibrium will remain. The forces P_1 & P_2 are, therefore, completely compensated for by filling the cylinders with water.

The effect of filling is not influenced by the shape of the vessel and only depends upon the area of the piston and the weight h below the surface.

Therefore, the unit pressure in a vertical direction is independent of the shape of the vessel; it is solely determined by the depth h below the surface and the specific gravity of the liquid.

As a matter of fact, the unit pressure at a given distance h below the surface of the liquid is the same in all directions.

---o---

HYDRO-DYNAMICS

Differential pressure as the Cause of Motion

In a quiet body of water, every particle, drop or molecule located in the same plane, is subjected to the same pressure.

$$p = h \lambda$$

as we have seen under Hydro-Statics.

If this condition is disturbed at any point, the particle next to it is subjected to a one-sided excess or differential pressure, and motion results affecting immediately the particles adjacent and also those remote.

Fig. 19 shows two reservoirs of different heights, filled with water. They can be connected with each other by opening a valve installed between the two. If this valve is opened, a pressure relief is effected in the first reservoir near and in the direction of the connection. As a result, the water will flow into the reservoir of lower pressure and over-flow.

By referring to Fig. 19, we see, that the pressures h_1 and h_2 are measured vertically above the reservoir connection. The excess pressure or pressure difference, which is causing the flow is expressed by the difference h between the two water surfaces.

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Stream or Flow Lines, Water Threads

The stream lines in which the particles travel have the tendency to be parallel to each other. All particles which flow in these stream lines, one behind the other, form what is called a Water Thread, Fig. 20.

---o---

Relation between Area, Velocity and Volume.

A particle of water is enclosed and fills completely the space f between two vertical planes of a pipe with constantly changing cross-sectional area. These planes are at right angles to the flow of water, see Fig. 21.

The two planes are the distance ds from each other and the particle moves during the time dt the distance of its own length ds.

The velocity of the water at this point is

$$v = \frac{ds}{dt}$$

Since the water fills the pipe completely, a new volume of the same magnitude must take the place of the one pushed ahead. Therefore, in the time dt a volume of

$$Q\,dt = f\,ds$$

passes said point.

Transposing this formula to

$$Q = f\,\frac{ds}{dt}$$ and substituting

v in place of $\frac{ds}{dt}$ we find

$$Q = f\,v \qquad (9)$$

Since the water cannot collect anyplace or form empty spaces, the same volume of water will have to pass in the same period of time through each plane or section of the pipe.

Consequently, if f_1 and f_2 represent different cross-sectional areas, Fig. 21, and v_1 and v_2 the velocities at these points, we have

$$Q = f\,v = f_1\,v_1 = f_2\,v_2 \quad \text{etc.} \quad \text{or}$$

$$\frac{v}{v_1} = \frac{f_1}{f} \quad ; \quad \frac{v_1}{v_2} = \frac{f_2}{f_1} \quad \text{etc.}$$

If a pipe is divided into certain equal volumes, the distances between these sections are proportional to the respective velocities of the water at these points.

---o---

INVESTIGATION OF A MERCURY COLUMN

Fig. 22 presents a device for measuring pounds pressure by the corresponding height of a mercury column, and forms the most accurate means of measuring pressure. A long vertical glass tube, A, open at both ends, rises out of a closed vessel B, in which mercury is placed, and the pressure to be measured is applied through C, to the surface of the mercury in the vessel. Before the pressure is applied, the level of the mercury inside and outside of the tube is the same as both are subject to atmospheric pressure. Now if pressure acts upon the surface of the mercury inside B but outside tube A, it is obvious that the pressures inside and outside tube A will immediately be unbalanced, and the mercury will rise in A.

From Letter No. 3, Sheet No. 3, it was found that a column of mercury 1 inch high exerted a pressure of .49 lbs. per square inch. Therefore, a pressure of one pound per sq. inch equals a column of mercury

$$\frac{1}{.49} \text{ inches or } 2.0408 \text{ inches high.}$$

This height of 2.0408 is measured above the surface of the mercury outside the tube. It is obvious that as some of the mercury is pressed out of the vessel B into tube A, the level of mercury in B will fall. This fall will be proportional to the quantity of mercury forced into the tube.

It would be incorrect, therefore, to measure off intervals of 2.0408" corresponding to each pound pressure, above the initial level of the mercury inside and outside the tube. Therefore, evidently a correction must be made for the decrease in the level in vessel B, or the mercury reservoir.

In order to keep the apparatus of a reasonable size, and to make the correction small, the tube A is made of relatively small cross-sectional area compared with the vessels.

We will now assume a case, and explain the calculation of the correction factor. Assume the inside diameter of tube A is 3/16" and of vessel B, 3".

With a pressure of one pound per sq. inch acting on the mercury in B the mercury will rise in A to a height of 2.0408". The area of A is .02761 square inches. Therefore, the volume of mercury pressed into A is 2.0408 x .02761 = .0563 cubic inches.

The zero on the scale of tube A should be coincident with the initial level of the mercury.

Now, the volume pressed into A has been taken away from the volume in B. The cross-sectional area of B is 7.0686 square inches.

The distance that the level in B has fallen in displacing .0563 cubic inches, is

$$\frac{.0563}{7.0686} \text{ or } .008 \text{ inches}$$

Therefore the correct distance to lay out above the zero mark, and corresponding to one pound pressure, is not 2.0408 inches, but 2.0408 inches minus .008 inches = 2.0328" and so on for each additional pound.

The total correction for 100 pounds would be .8 inches. If we took a tape line and measured the distance between the zero and the mark corresponding to 100 pounds we would find it to be 204.08 inches minus .8 inches or, 203.28".

---o---

It may be wondered why mercury is used in gauges, as above described. Mercury is the heaviest of all the liquids. Any other liquid could be used as a medium of measurements, water for instance, but an exceedingly bulky gauge would result. One pound pressure, for instance, equals a column of mercury only 2.0408" in height, while the equivalent height of a column of water is 2.31 feet. 100 pounds pressure therefore, which is very common in pump practice, would have to be measured by a water column 231 ft. high, which is obviously impractical.

The equivalent height of a column of liquid to correspond to a given pressure varies inversely as the specific gravity of the liquid. That is, the heavier the liquid, the lower the column and vice versa. Therefore, the minimum height of the column will correspond to the heaviest liquid, Mercury, which results in the smallest and most compact liquid gauge.

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VELOCITY OF EFFLUX AND ACTUAL FLOW THROUGH AN ORIFICE, FLOW THROUGH PIPES OF VARIABLE CROSS SECTION, THEOREM OF BERNOULLI

Velocity of Efflux

The whole modern science of hydraulics is built on the well known fundamental formula

$$v = \sqrt{2gh} \qquad (4)$$

This is the same as the fundamental formula in Letter 2.

While this formula, as applied to hydraulics, was first published by John Bernoulli (Bern, Switzerland) in 1732, the law which it represents was discovered long before that date.

In 1640, Castelli, a pupil of Galileo, published his observations on a waterclock bowl; he had found that more water was discharged in a given time when the bowl was full than when it was nearly empty.

Only a few years later, in 1644, Toricelli proved that the discharge velocity from an orifice in the wall of a vessel varies as the square root of the head of the water on the orifice, and

$$v = \sqrt{2gh}$$

is now known as the Toricelli theorem.

Fig. 23 shows a vessel filled with water, and having a small hole a in one of its sides at a distance h below the surface. The water flows through this opening with a certain velocity which we call v.

In the following explanation, as in all previous and most following chapters, friction, viscosity and similar losses are not taken into consideration. In practical hydraulics, such losses, of course, play an important part, and many experimental values can be found in hydraulic literature.

The proof of Toricelli theorem is as follows:

From letter 7, we found that the formula for Potential energy is

$$E_p = w\,s \qquad (22)$$

For a mass m, at the height h, this may be written

$$E_p = m\,g\,h$$

In order to pass through the orifice a the mass m has to fall the distance h, and at this point its entire potential energy will have been transformed to kinetic energy.

$$E_k = \frac{m\,v^2}{2} \qquad (23)$$

Since, disregarding friction losses, no energy is lost, its E_k when emerging from the orifice must be equal to its E_p at height h, we have

$$m\,g\,h = \frac{m\,v^2}{2}$$

In this equation m cancels out, and we have

$$\frac{v^2}{2} = g\,h, \text{ or}$$

$$v^2 = 2\,g\,h, \text{ or}$$

$$v = \sqrt{2\,g\,h} \qquad (24)$$

---o---

ACTUAL FLOW OF WATER THROUGH AN OPENING OR ORIFICE

It will be both advantageous and of interest to investigate the actual flow of water through an orifice, and the so-called coefficients governing such flow. These are the velocity and the contraction coefficients, and their product which is called the discharge coefficient.

VELOCITY COEFFICIENT

When comparing the capacity that actually flows through an orifice with the volume that should theoretically flow through it, we find that the actual volume is less than the theoretical, in other words

$$\frac{\text{Actual Velocity of Flow}}{\sqrt{2\,g\,h}} = \text{Velocity Coefficient} < 1$$

(< stands for smaller than one)
(> stands for larger than one)

This is not at all surprising, when we remember that our calculations heretofore have been based on frictionless motion, viz: transforming potential into kinetic energy without losses, such as in our proof of the Toricelli Theorem.

Such supposition is not true. Friction occurs between the water and the containing walls, also between the water particles themselves.

A part of the potential energy is consumed in overcoming this friction, and is apparently lost. In reality it is not lost at all, but has changed into heat. However, as such heat is useless in hydraulics, we must consider this wasted energy.

On account of the loss through friction, the exit velocity must be less than the theoretical, and consequently the velocity coefficient must be less than unity.

If φ represents the relation between actual and theoretical velocities, then the actual exit velocity will be

$$v = \varphi \sqrt{2\,g\,h} \qquad (27)$$

φ is the Velocity Coefficient.

The actual "capacity" discharged bears the same ratio to the theoretical, as the actual "velocity" bears to the theoretical.

CONTRACTION COEFFICIENT.

In addition to friction, the actual volume discharged is decreased as compared with the theoretical due to another cause, and this is the shape of the discharge orifice.

Referring to Fig. 24, if the orifice is so shaped that the outside water threads converge as they flow out, they crowd one another towards the center of the issuing jet, and the stream is contracted, outside the orifice. The quantity of discharge, therefore, does not depend upon the orifice of area f, but upon the area of the stream f_1, outside of the orifice.

If $\frac{f_1}{f} = \varepsilon$ then the exit stream area

$f_1 = \varepsilon f$ where

ε = the Coefficient of Contraction.

---o---

COEFFICIENT OF DISCHARGE

From the foregoing it follows that the actual volume, instead of being

$$Q = f v$$

is

$$Q = f \varepsilon \varphi \sqrt{2 g h}$$

The product of ε and φ is called ζ, which is the coefficient of discharge. Then

$$Q = \zeta f \sqrt{2 g h}$$

This coefficient ζ represents a loss of energy, and a knowledge of it is important. If ε and φ are desired separately, at least one has to be determined directly. It is possible to measure the contraction by measuring the diameter of the stream at f_1, but this process is complicated and not satisfactory. Therefore, it is the usual practice to determine ζ by measuring the actual quantity discharged, and comparing it with the theoretical. In this manner, a series of constants is secured from a number of experiments, and these are used in solving problems in hydraulics.

CONTRACTION WITH ORIFICES OF DIFFERENT SHAPE

Refer to figures 24, 25, 26 and 27. The contraction as before stated, depends on the shape of the orifice. With reference to the four types illustrated, it is evident that contraction is greatest with Fig. 25.

Less contraction occurs with Fig. 24, which has sharpened inside edges, and is known as an "orifice in a thin plate"

With Fig. 26, the contraction still further diminishes with the coverging angle δ.

At Fig. 27, with rounded edges, the contraction disappears entirely, and the contraction coefficient becomes unity. The contraction coefficient disappearing, the velocity and discharge coefficients become one and the same thing.

If the orifice is circular, the stream remains solid even under sharp contraction. However, any other shape of orifice results in a disturbance of motion, and in losses of energy, easily seen in the broom-like shape of the different water threads, and the general whirling and eddying of the issuing stream.

Only orifices shaped like Fig. 27, give a nearly solid, smooth stream, the water threads issuing parallel to one another, but even then the circular orifice gives the most compact and evenly flowing stream.

Wherever contraction is present, the coefficient changes, as experiments have demonstrated, with the head or pressure acting upon the orifice. The discharge coefficient when dealing with contracted orifices is therefore not very well adapted for calculating capacities for other but similar orifices under different heads.

The velocity coefficient, on the other hand, is nearly independent of the head on the orifices.

Circular orifices of the shape in Fig. 27, that is, without contraction - are used in measuring quantities of water. As in this case, the velocity and discharge coefficient are the same thing, and consequently the discharge coefficient is independent of the pressure, the actual capacity discharged can be very definitely predicted.

For a circular orifice like Fig. 27, the discharge coefficient varies according to the smoothness of the orifice walls.

It is important that the parallel walls of this kind of orifice be no longer than necessary, but just long enough to insure parallel flow of the different water threads. The rounded edges of the approach do not have to be of large radius.

Experiments by Hansen have resulted in values of ζ ranging from 0.9938 to 0.9986, for nozzles designed and built along the lines just described.

FLOW THROUGH PIPES OF VARIABLE CROSS-SECTION

This discussion is a forerunner to the important Bernoulli Theorem, a complete understanding of which is necessary in the building up of the Main Equation of the Centrifugal Pump.

Fig. 28 shows a tube or pipe of variable cross-section with piezometer (small upright) tubes at different points.

A constant quantity of water is flowing in the direction of the arrow and the tube is supposed to be filled completely with water. The water level of each piezometer tube shows the pressure exerted by the liquid upon the walls at each point from 1 to 5 and the vertical distance between the water level and the center of the pipe represents the magnitude. At the first glance it might seem puzzling that the pressure is greatest at the widest parts (1 and 5) and the smallest at point 3.

It is easily seen, however, that the velocity of the passing water must be highest at the smallest area and vice versa.

Assuming a steady constant flow, the water must undergo a change of velocity, in order to flow from a larger cross section through a smaller cross section in the same length of time. In order to produce this acceleration there must be acting a greater pressure from behind; therefore the pressure at point 1 is greater than at point 2, and the pressure at point 2 is greater than at point 3.

Similarly, in passing from the smaller to the larger part of the tube the velocity decreases, or the acceleration is negative; hence the pressure is greater ahead than behind.

This shows conclusively and very clearly that the pressure in a pipe line is greatest at the largest part, and the velocity smallest.

A following letter will explain this still further.

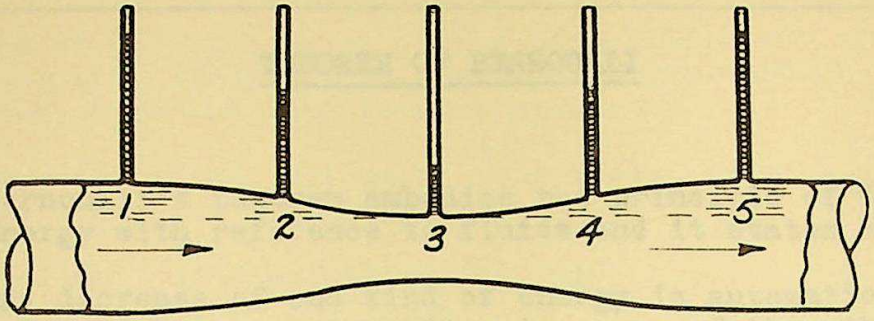

FIG. 28

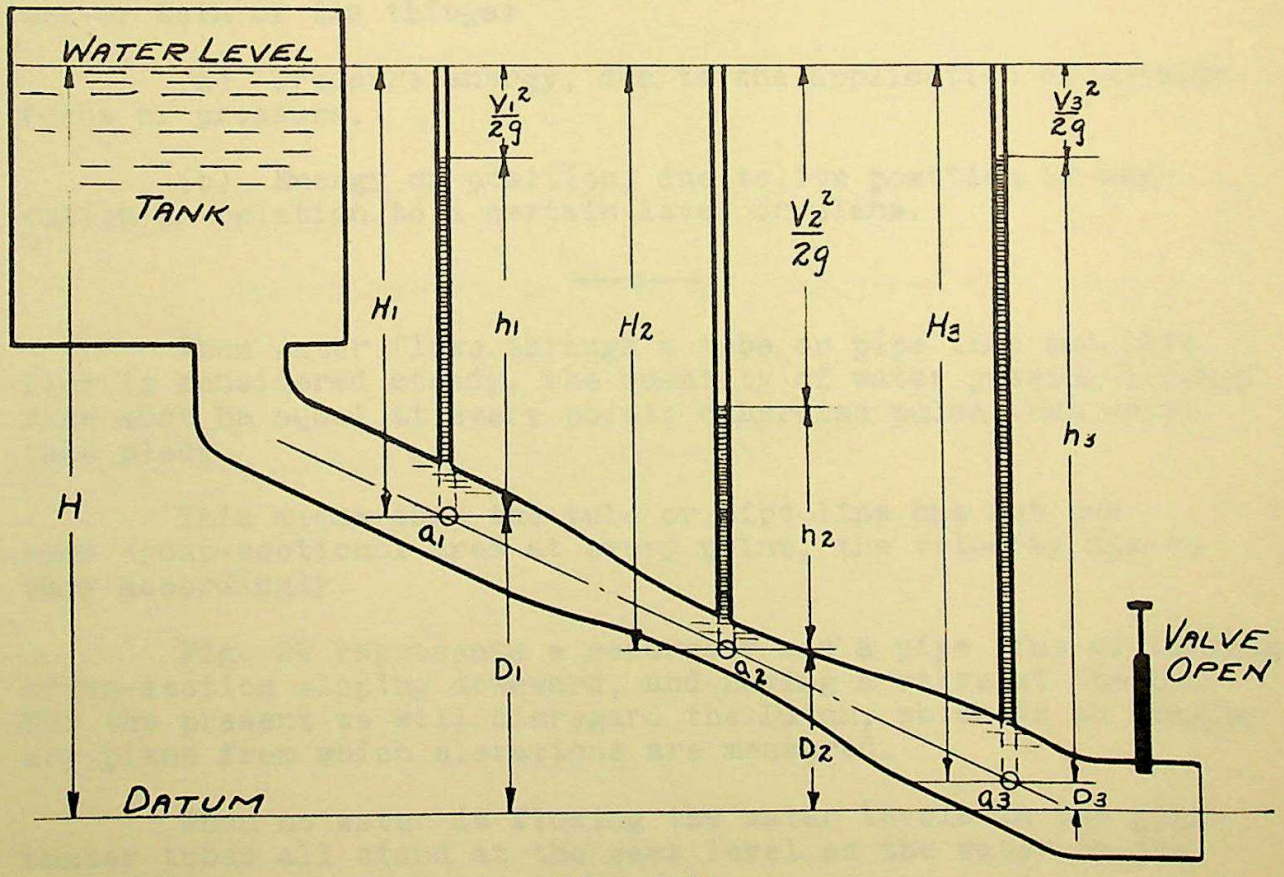

FIG. 29

PLATE NO. 9

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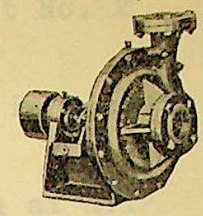


CENTRIFUGAL PUMPS - THEIR THEORY AND APPLICATION

Letter No. 9

THEOREM OF BERNOULLI

Bernoulli's theorem embodies the principle of Conservation of Energy with reference to fluids and it states that:

Any decrease of one kind of energy is automatically counter-balanced by a corresponding increase of the other.

There are two kinds of energy, as explained in Letter 7.

(1) Potential (2) Kinetic

In hydraulics, potential energy may be due to either one or both of two things:

(a) Pressure energy, due to the application of outside force or pressure.

(b) Energy of position, due to its position or elevation in relation to a certain level or plane.

---o---

When water flows through a tube or pipe line and this flow is considered steady, the quantity of water passing through same must be equal at every point; otherwise pulsations would take place.

This means that the tube or pipe line has not the same cross-sectional area at every point, the velocity has to vary accordingly.

Fig. 29 represents a reservoir and a pipe line of varying cross-section sloping downward, and having a valve at the end. For the present we will disregard the Datum, which is an imaginary plane from which elevations are measured.

When no water is flowing the water levels in the piezometer tubes all stand at the same level as the water in the reservoir.

Pressure gauges at points a_1, a_2, etc. (the points of connection of the piezometer tubes to the pipe line) would show pressure equal to the static difference between the point of connection and the reservoir level. (H_1, H_2, etc.)

As soon as the valve is opened the water is set in motion by gravity.

The piezometer tubes now show the water standing at different levels.

The drop in the level of the piezometer tube simply represents the head necessary to create the flow through the area at the point where the tube is attached.

At point a_1 the height h_1 represents the remaining static head or pressure head at that point.

Similarly with points a_2, a_3, etc.

By introducing the weight w of water flowing through the system in each second (the same as in the Toricelli theorem) we obtain terms of energy.

The student will notice that in most of the future illustrations and analyses of equations the principle of work is embodied.

The potential energy was explained in letter 7 as

$$E_p = w\ s \qquad (22)$$

s represented distance, and in hydraulics we substitute the height h for it, similar to equation 4.

We therefore have at point a_1 when water is at rest

$$E_p = w\ H_1 \quad \text{(foot pounds)}$$

When the water is in motion the total energy consists: the pressure energy $w\ h_1$ (see figure 29)

plus the energy of the motion, kinetic energy

$$E_k = \frac{m\ v_1^2}{2} \qquad (23)$$

By substituting $\frac{w}{g}$ for m we have

$$E_k = \frac{w}{g} \cdot \frac{v_1^2}{2} = \frac{w\ v_1^2}{2\ g}$$

We therefore, have at point a_1, when water is moving

$$E_k = \frac{w\ v_1^2}{2\ g}$$

It follows now that

Total Energy = Pressure Energy plus Energy of Motion

$$w H_1 = w h_1 + \frac{w v_1^2}{2 g}$$

or $$H_1 = h_1 + \frac{v_1^2}{2 g} \qquad (28)$$

Applying similar reasoning to point a_2 we have

$$H_2 = h_2 + \frac{v_2^2}{2 g}$$

---o---

It is seen that the original total static head has decreased and the balance has been converted into dynamic head. That is, the original pressure head H_1 is now split up into two parts, the remaining pressure head h_1 and the difference, which is consumed in producing the velocity of the water at point a_1.

This is $\frac{v_1^2}{2 g}$

and is called "Velocity Head". A moment's thought will make it evident that the higher the velocity, the greater the Velocity head will be, and the smaller the Pressure head will be; and vice versa.

If points a_1 and a_2, for instance, are on the same level, (similar to Fig. 28), it is evident that the total pressure heads, H_1 and H_2 will be equal. From this it follows that

$$h_1 + \frac{v_1^2}{2 g} = h_2 + \frac{v_2^2}{2 g}$$

That is, the sum of the velocity and pressure heads in a pipe, through which continuous steady flow is taking place, is equal at various points. In other words, the sum of these two heads (the total head) is constant. Friction, of course, is neglected, and, as before stated, the pipe is assumed to be level.

This law may be roughly stated: the higher the velocity, the lower the pressure, and vice versa.

Now returning to the sloping pipe, Fig. 29, let us introduce the Datum (an imaginary plane) as shown. Points a_1, a_2, and a_3 are located the vertical distances D_1, D_2, and D_3 above the Datum and the surface of the water in the reservoir.

It is evident that all points in the Datum will be under the same total head H, but the total head for a point in the Datum under a_1, may be considered made up of

$$D_1 + h_1 + \frac{v_1^2}{2g}$$

Similarly, for a point in the Datum under a_2 the total head is

$$D_2 + h_2 + \frac{v_2^2}{2g}$$

As these are all equal, we may write

$$H = \frac{v_1^2}{2g} + h_1 + D_1 = \frac{v_2^2}{2g} + h_2 + D_2 = \frac{v_3^2}{2g} + h_3 + D_3 \text{ etc.}$$

At point a_1 $\frac{v_1^2}{2g}$ is the velocity head

h_1 is the pressure head

D_1 is the potential head, or height above an arbitrary level.

Bernoulli's Theorem may now be stated in full as follows:

IN A PIPE OF VARIABLE CROSS-SECTION, AND WITHOUT ABRUPT BENDS OR TURNS, WHEREIN STEADY CONTINUOUS FLOW IS TAKING PLACE THE SUM OF THE VELOCITY PRESSURE AND POTENTIAL HEADS AT ONE POINT, NEGLECTING FRICTION, IS EQUAL TO THEIR SUM AT ANY OTHER POINT. IN OTHER WORDS, THE SUM OF THESE THREE HEADS IS ALWAYS A CONSTANT.

---o---

A practical illustration of the application of Bernoulli's Theorem is the Venturi Meter, one of the most accurate instruments for measuring water.

PERFORMANCE WHEN DISCHARGING THROUGH DIFFERENT PIPE LINES

In Fig. 30 the Q-H Curve, and the Head of System (40 ft. Static) Curve are the same as those we have been considering. Let us assume the pumping requirements are 400 G.P.M. through one pipe line against 40 ft. Static Head.

Let us also assume that there are two other discharge lines, and that the pump is required to pump through any one, but only one, of these lines at a time.

The second pipe line runs to an elevation 50 ft. above the level of the pump, consequently the static head is 50 ft. The size and length of pipe is such that the friction for 400 G.P.M. is 30 ft. That is, for 400 G.P.M. the total head is 80 ft. The capacity delivered through this at constant speed of 1740 R.P.M. is determined as follows:

Draw horizontal line 50 ft. Static, and lay off the Head of System Curve above it as a base.

The intersection of this Head of System Curve with the Q-H Curve gives us the desired capacity, about 347 G.P.M.

---o---

The third pipe line extends to an elevation 15 ft. above the pump, but the friction for 400 G.P.M. is 35 ft. How much water will be delivered?

Proceeding in similar fashion, and plotting a new Head of System Curve for 15 ft. Static head, we obtain, at the intersection, a capacity of about 442 G.P.M.

---o---

SEVERAL PUMPS DISCHARGING THROUGH ONE PIPE LINE AT THE SAME TIME

Suppose two pumps are installed, one only to be operated under normal conditions, which are 400 G.P.M. against 66 ft., this head composed of 40 ft. static and 26 ft. friction. If an emergency arises, such as a conflagration, what will the combined capacity be if the two pumps are operated at the same time? The natural answer would be, double the capacity of one pump, or 800 G.P.M. This is not correct, as will be shown. Referring to Fig. 31 the Q-H curve for two pumps is a "composite" of the two pumps, which are supposed to be duplicates. That is, for a given head, the capacity is double that of a single pump. The point where this composite Q-H curve intersects the "Head of System" Curve A determines the total capacity that will be delivered, in this case approximately 510 G.P.M.

It is thus seen that the capacity with two pumps operating is only 27% more than that when only one pump is in service. The increase in pipe friction increases the total head so much that the total capacity is greatly diminished. If it is desired to have the total capacity with two pumps running more nearly double than when only one pump is operating, it is necessary to use a larger pipe line, so that the friction will not form so great a percentage of the total head.

---o---

Suppose 800 G.P.M. are to be delivered against 40 ft. Static Head through a pipe line, the friction of which is 26 ft. for this capacity. Two duplicate pumps are installed, each designed for 400 G.P.M., 66 ft. Total Head. How much water will be delivered if only one pump operates?

Referring to Fig. 31 lay off the Q-H curve for one pump, which is the same as that for two pumps, except for each head only one-half capacity is delivered. Also lay off the Head of System Curve B. The point of intersection determines the capacity of one pump. It is approximately 480 G.P.M.

It is thus seen that when only one pump is operating, the capacity is considerably more than half the combined capacities.

---o---

The problem often arises, how will 3 pumps work in parallel (operating simultaneously) on one pipe line, what will the combined capacity be, etc.

In order to predict the performance it is necessary to know the Q-H Curve of each pump.

Referring to Fig. 32, Curves A, B, C are the characteristics of the three pumps in question. It will be noted that they all have different shut-offs, but they are all pumps of the same class, that is, moderate head pumps. Obviously, this should be so. Very low head, and high head pumps would not work satisfactorily together. The high head units would build up a pressure sufficient to close the check valve in the discharge from the low-head pump, and it would not deliver any water at all.

Continuing the problem, E is the head curve of the piping system, and the static head is 30 ft. Pumps A, B, and C when operating <u>singly</u> will deliver 475, 285 and 650 G.P.M. through the line respectively.

It might be assumed that when all three units are working at the same time, the combined output would be the sum of the separate capacities or 1410 G.P.M., but this is not so.

The increased capacity when the three pumps are running builds up the total head due to the greater pipe friction which results in the total capacity being less than might otherwise be expected.

Curve D is a composite Q-H Curve. It represents the sum of the capacities of the three pumps. It is obtained by reading off the capacities of A, B, and C, at 80 ft. head, for instance, adding them together and plotting the point. This is 245, 120 and 280 and the sum is 645. Similarly for 70, 60, 50 ft. heads, etc., the combined capacities are 955, 1150 and 1315 G.P.M. respectively.

The point where Curve D intersects Curve E determines the combined capacity of the pumps, namely 990 G.P.M.

---o---

The combined capacity of any two pumps could be determined by plotting a composite Q-H Curve from their individual characteristics, and noting where it cut Curve E.

In this letter we have given several examples of the practical use of characteristic curves. They represent the best method of studying and applying centrifugal pumps to pumping problems.

It should be remembered, however, that as pipe friction is a variable quantity, the Head of System Curves must not be taken too literally.

---o---

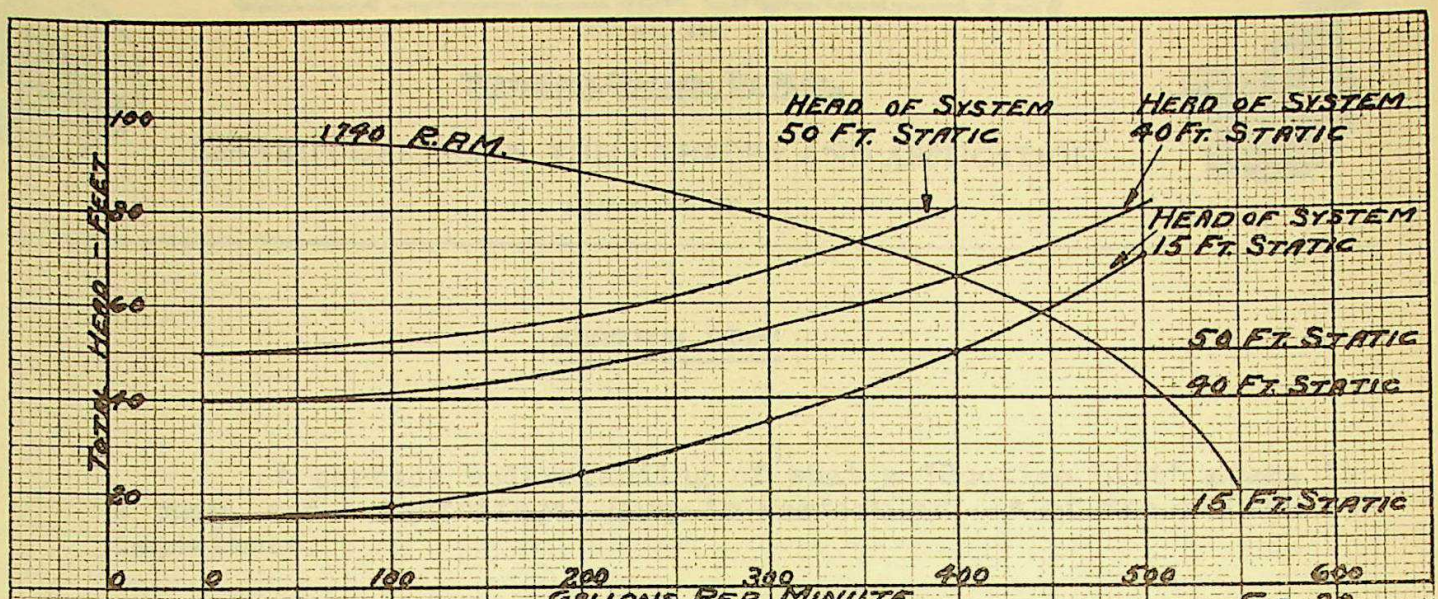

1740 R.P.M.
HEAD OF SYSTEM 50 FT. STATIC
HEAD OF SYSTEM 40 FT. STATIC
HEAD OF SYSTEM 15 FT. STATIC
50 FT. STATIC
40 FT. STATIC
15 FT. STATIC
TOTAL HEAD - FEET
GALLONS PER MINUTE

FIG. 30

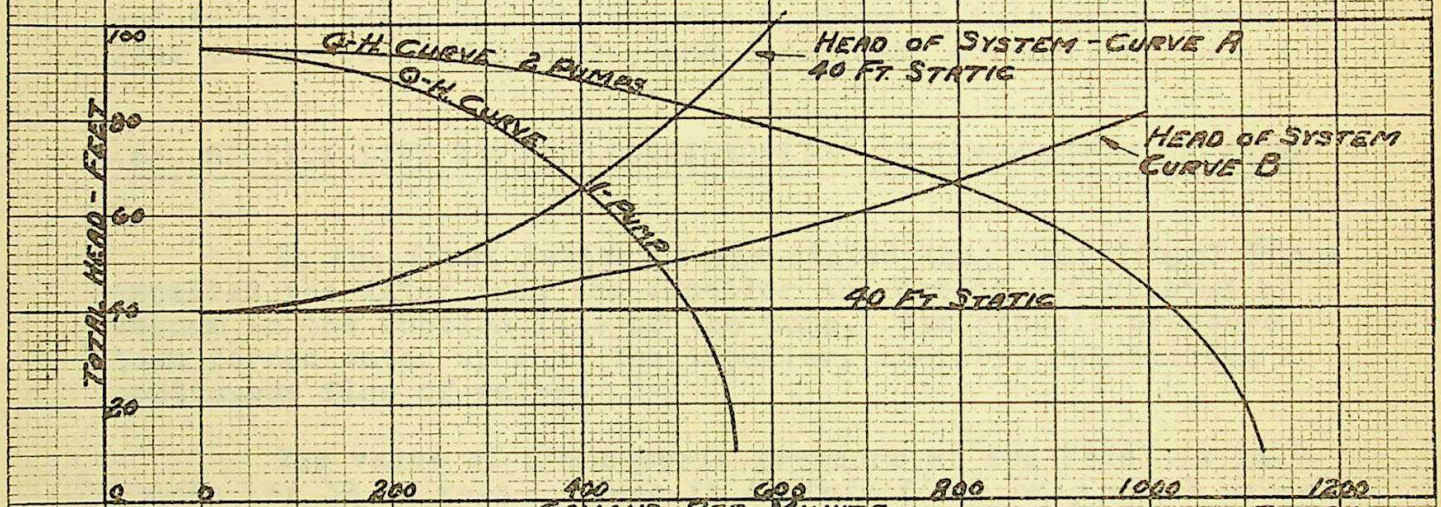

Q-H CURVE 2 PUMPS
Q-H CURVE 1-PUMP
HEAD OF SYSTEM - CURVE A 40 FT. STATIC
HEAD OF SYSTEM CURVE B
40 FT. STATIC
TOTAL HEAD - FEET
GALLONS PER MINUTE

FIG. 31

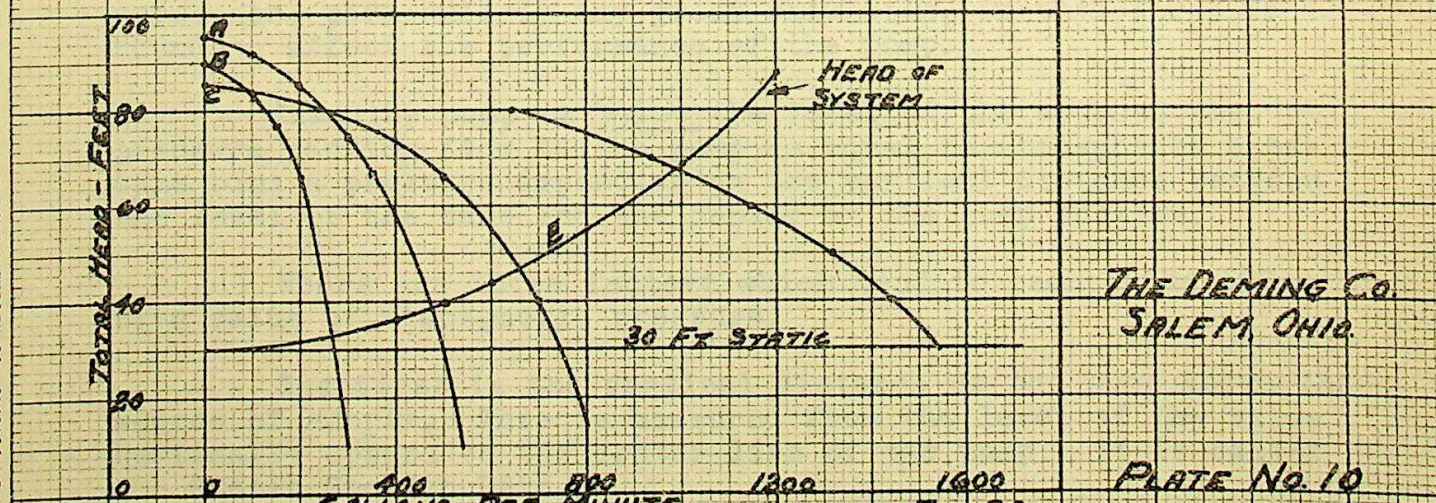


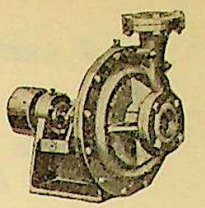
FIG. 32

THE DEMING CO.
SALEM, OHIO.

PLATE NO. 10

EUGENE DIETZGEN CO., CHICAGO-NEW YORK NO. 346 B

THE DEMING COMPANY
ESTABLISHED 1880
SALEM, OHIO, U.S.A.



CENTRIFUGAL PUMPS - THEIR THEORY AND APPLICATION

Letter No. 10

SUCTION LIFT

A correct understanding of what a "Suction Lift" means to a pump, and a Centrifugal Pump in particular, is of the utmost importance.

If a centrifugal pump does not function as expected, connect a vacuum gauge to the tapped opening of the suction nozzle and in nine cases out of ten, the reading of this gauge gives the clue to the seat of trouble, and consequent correction.

(A) SUCTION LIFT WITHOUT CONSIDERATION OF TEMPERATURE OF WATER

In the first letter it was stated that Centrifugal Force expelled the water from the impeller and casing and created a vacuum in the suction line of the pump. Atmospheric pressure then forced water up into the impeller, thereby, maintaining a continuous flow of water.

The value of atmospheric pressure at sea level is 14.7 pounds, or 33.96 feet. Since it is this force only which causes water to flow to the impeller, assuming that the pump is above the level of the source of supply, it is quite evident that any reduction, particularly any great reduction, in this force is bound to affect the performance of the pump.

When the pump is below the level of the source, the pressure causing flow to the pump is equal to atmospheric pressure plus a pressure depending on the vertical distance between the level of the same and the center of the pump suction.

There are three factors which tend to reduce the effective value of atmospheric pressure.

The first is the vertical distance of the pump above the source of supply. This is measured from the level of the water vertically upward to the center line of the pump. It is called the Static Suction Lift.

The second factor is known as Velocity Head and depends on the velocity of the water in the suction pipe. It is numerically equal to

$$\frac{V_s^2}{2g} \qquad (24)$$

where

V_s = Velocity of water in the pump suction

g = Acceleration due to gravity = 32.2 feet per sec. per sec.

The third is known as Friction Head and depends on the suction velocity, on the length of the suction pipe and on the number and kind of fittings in the suction line.

The pressure at the suction connection of the pump is

$$P_s = -h_s - \frac{V_s^2}{2g} - h_f \qquad (25)$$

where

P_s = Pressure at suction connection of pump

$-h_s$ = Static suction lift

$+Z_s$ = Static suction head

$\frac{V_s^2}{2g}$ = Velocity head

h_f = Friction head

Since all these factors have minus signs, their sum is negative and, therefore, less than atmospheric pressure. If this pressure is subtracted from atmospheric pressure, the absolute pressure tending to cause flow of water to the pump is obtained.

The above formula is a simple application of the Bernoulli Theorem, fully explained in the preceding letter.

When the pump is below the source of supply, h_s takes a plus sign and P_s may be either negative or positive, depending on which has the greater value, h_s or the sum of the other two factors.

(B) SUCTION LIFT AT VARIOUS TEMPERATURES OF WATER

The lowest possible absolute pressure in a vessel, filled with liquid (water), is the Vapor Pressure corresponding to or caused by the temperature of the liquid.

Values of these vapor pressures are tabulated in so-called steam tables.

When these pressures are present the water begins to steam which separates the water from the walls of the vessel (cavitation) with its detrimental effects upon the various materials of pumps. Therefore, we may restate the above as follows:

"The maximum possible Suction Lift is reached, when the pressure at any point in the suction line drops to the Vapor pressure corresponding to the temperature of water being pumped. The pump 'breaks suction'."

The foregoing explanation is combined into the following formula:

$$h_{s\ max} = A - (V_t + \Sigma h_f + \frac{V_s^2}{2g} + \Delta h) \qquad (30)$$

where

$h_{s\ max}$ = Maximum possible suction lift.

A = Atmospheric pressure in Ft. per sq. inch.

V_t = Vapor pressure of water in Ft. per sq. inch at the respective temperature.

Σh_f = Sum of head losses in suction line from foot valve to suction nozzle of pump, due to friction, change of areas, etc., in feet.

$\frac{V_s^2}{2g}$ = Velocity head in feet.

Δh = 10.5 ft. A constant, given by C. Pfleiderer in "Die Kreiselpumpen." This keeps the pressure at the suction of the pump 10.5 ft. greater than the boiling point of water and is a safety factor since there are additional hydraulic losses from the suction flange of the pump to the impeller inlet.

All the above units can be expressed in pounds, if desired.

ILLUSTRATION:

(a) What is the maximum static suction lift of a pump placed at sea level where the atmospheric pressure is 14.70# per square inch?

The temperature of the water is 126.3° F, which corresponds to an absolute pressure of Vapor V_t = 2#. (Fig. 34, Plate 11)

The sum of the head losses in suction line, from foot valve to the pump nozzle, are Σh_f = 10 ft. = 4.33#.

The velocity head is assumed as $\frac{V_s^2}{2g}$ = 2.25 ft. = .975#

Δh = 10.5 as stated above

We now have as per formula (30)

$h_{s\ max}$ = 14.7 - (2 + 4.33 + .975 +4.55) = 14.7 - 11855
= 2.845# = 6.6 ft.

---o---

(b) If the pump is placed 1 mile above sea level where the atmospheric pressure is 12.02# per sq. in., with temperature of water and all other values remaining the same, we find

$h_{s\ max}$ = 12.02 - 11.855 = .165# = .38 ft.

---o---

(c) Increasing the temperature to 202°, with pump 1 mile above sea level and all other factors remaining the same

$h_{s\ max}$ = 12.02 - (12 + 4.33 +.975 +4.55) = 12.02 - 21.855
= -9.835# = -22.71 ft.

This means that the pump has to be placed under a suction head of at least 22.71 ft. for satisfactory performance.

---o---

The term "Vapor Pressure" possibly needs a little further explanation.

If water is heated to a temperature of 212° F. (water boils in the kettle) and the steam rises from the surface, the water is changing into steam.

We know from Letter 3, Page 1, that the atmosphere surrounding us exerts a pressure of 14.7# per sq. in. upon everything. This pressure is also present upon the surface of the water in the kettle and, therefore, the vapor pressure must be equivalent to the atmospheric pressure of 14.7# in order to be released from the water.

At 212° F. the absolute pressure of vapor is 14.7#.

Scientific research has established so-called Steam Tables which show that with lower surrounding pressures the vapor pressures also decrease. (See steam tables, Plate 11, Fig. 34.)

If water is placed in a vessel where different vacua (less than atmospheric pressure) can be obtained, water will vaporize (boil) at lower temperatures; for instance, in a vacuum of 9.56" at a temperature of 193.2° F.

Applied to a pump, suppose an impeller creates a vacuum of 9.56" and the temperature of the water is 193.2° F.. From the Steam Tables we note that at this temperature the water vaporizes (boils) and since the ordinary centrifugal pump cannot handle vapor, it breaks suction.

On Plate 12, Fig. 36 the "Theoretical Suction Lift" curve for water at various temperatures together with a "Practical Suction Lift" curve, is based on experiments and formula of C. Pfleiderer. This latter curve deviates greatly from the conventional "Practical Suction Lift" curve as found in various publications and pump catalogs and for comparison is shown in dotted lines. As far as we can see this curve was simply established by someone drawing a straight line - and making a good or a bad guess. The customary lift of 15 ft. as a maximum is not adhered to in present day installations, although it is common practice of pump manufacturers not to guarantee pumps working under more than 15 ft. total suction lift. By using the curve (Fig. 36) in connection with formula (30) on Page 3 of this letter, absolutely safe results will be obtained with the average pump.

For easy reference Plate 11 has all necessary tabulations, with the exception of pipe friction tables (given in Deming catalogs) to figure maximum suction lifts easily and correctly.

INFLUENCE OF SUCTION LIFTS UPON DELIVERY AND PUMP EFFICIENCY

Moderate suction lifts can be handled just as readily with a centrifugal as with a displacement pump, provided the suction line is properly proportioned to the capacity of the pump, absolutely air tight and properly laid so that air pockets cannot form.

Suction lifts approaching and above 20 ft. can be successfully handled but a good knowledge of the controlling factors is necessary in selecting such pumps.

The curves given on Plate 12, Fig. 37 represents actual tests of a Deming Fig. 4010 #1½ at a speed of 3500 R.P.M., under suction lifts of 0, 15', 25' and 30'. It will be noticed that up to 15 ft. the suction lift does not have any detrimental effect upon efficiency nor does the Capacity-Head curve change. With a suction lift of 20 ft. the capacity head curve falls off beyond the 120 G.P.M. delivery, but from the Zero capacity to 120 G.P.M. efficiency, as well as capacity head are the same as under a 15 ft. Suction Lift . With 25 ft. suction lift, reduction in head and efficiency are quite pronounced beyond 60 G.P.M. The pump was not objectionably noisy under 25 ft. suction lift although the rattling noise (cavitation) was present on capacities larger than 60 G.P.M.

In a lift of 30 ft. the performance was unstable and pump ran very noisy.

If a pump has to work under a considerable suction lift, select a pump where the quantity required lays to the left of its best efficiency on the conventional type of characteristic curve. For instance, in our illustration 110 G.P.M. is towards the left of the maximum efficiency point and therefore suitable for a higher suction lift than if 135 G.P.M. were required. For this larger capacity the next size larger pump should be selected for high suction lifts.

It is obvious that the suction passage to the impeller and the impeller itself control the maximum suction lift which a centrifugal pump can effectively deal with. Various means are available to the skillful designer to accomplish this end.

Besides the inlet opening of the impeller and its speed, the ratio of outside to inside diameter of the impeller itself is an important factor. This ratio to a great extent governs the efficiency as well as suction ability and probably best results are obtained with impellers which have a ratio of 1 to 2 up to 1 to 2.5; For instance, the impeller inlet opening or eye is 3" then the outside diameter should be between 6" and 7½". If this proportion is exceeded the disc friction becomes a handicapping factor, as far as efficiency is concerned, and if on the other hand this ratio is decreased, the impeller vanes become too short and the suction lift which impeller can produce is impaired. This is one of the reasons why for high speeds and comparatively low heads, the helicoidal type impeller is used, which type gives longer vanes for a given ratio between impeller eye and outside diameter.

In conclusion attention is called to pump installations where the total head is entirely, or almost so, a suction lift. Such conditions are encountered a good deal on construction work, pumping out ditches, coffer dams, etc., where it is simply necessary to lift the water up to the pump and discharge it over a bank, into a flume, etc.

Installations of this kind are liable to cause trouble, pumps easily lose their suction by pulling air through the discharge nozzle of the pump.

As an illustration, assume that a pump has been designed to deliver its rated capacity against a total head of 25 ft. and it is installed so that the actual total suction lift is 20 ft., which means that at the impeller inlet a vacuum of 17.5" mercury exists and at the impeller periphery there is only the remaining head of 5 ft. or 2.17# pressure present above the atmosphere of 14.7#. With this 2.17# excess pressure the water between impeller and pump outlet has to prevent air from short circuiting through the pump discharge into the impeller eye. It is a well known fact that the impeller passages are very seldom completely filled with water, there are eddie currents, vacua or air spaces. Also in the casing or volute of the pump an even pressure does not exist at all points. Especially near the so called throat of the volute quite frequently a vacuum has been observed. This immediately brings the danger

point close to the pump discharge nozzle, and with a horizontal top discharge and a horizontal piece of discharge pipe a direct communication between impeller inlet and atmosphere is established, should the pipe line happen to be not completely filled with water.

To prevent this condition it is the general practice of pump manufacturers to recommend a 3, 4 or 5 ft. section of vertical discharge pipe forming an effective water seal.

Suction lifts of centrifugal pumps are a subject which has the careful attention of the designer, not so much on small, every day pumps but on the larger pumps.

The Hydraulic Institute, of which we are a member has lately established the so-called "Specific Speed" as the basis for determining the maximum suction lifts for different conditions and when pumps are selected in accordance with the same, so-called cavitation resulting in noise and corrosion are avoided.

A great many factors enter into the cause for this phenomenon, but such discussions would us put beyond the scope of this treatise, and be of little interest or help to those interested in our lessons.

PERFORMANCE OF CENTRIFUGAL PUMPS CONNECTED IN SERIES

Very often the question arises: How will two pumps of different Capacity-Head characteristics act when connected in series, i.e. the first pump discharging into the suction of the second pump.

In order to demonstrate this in a simple and clear manner, we have conducted four tests and the results are given on Plate #13, Figs. 38, 39 and 40.

1) Deming Side Suction Pump, Fig. 4010 #2 operating at 1750 R.P.M. (Low head).
2) Deming Side Suction Pump, Fig. 4010 #2 operating at 3500 R.P.M. (High head).
3) Low speed pump discharging into suction of high speed pump.
(4 High speed pump discharging into suction of low speed pump.

Fig. 38, lower position, shows the Capacity - Head curve at 1750 R.P.M. and it will be noted that the total head produced at Shut-off or No-delivery is 40 ft. and the maximum capacity at no head is 155 G.P.M.

Fig. 38, upper position shows the Capacity - Head curve of the same pump at a speed of 3500 R.P.M.

Fig. #39 represents the high speed pump delivering into the low speed pump. Up to the maximum capacity of the slow speed pump the performance is what would be expected, namely the total

pressures at the various points are the sums of the individual points. But beyond the 155 G.P.M. point the head obtained at the discharge nozzle of the second pump is less than the head which the high speed pump is able to create alone. In other words, there occurs a considerable friction loss when forcing more water through the slow speed pump than the maximum capacity for which it is designed.

Fig. 40 gives the results obtained when the discharge line of the slow speed pump is connected into the suction of the high speed pump, measuring the pressure at the discharge nozzle of the second pump.

The total head obtained at no delivery (shut-off) is approximately that of the first pump plus that of the second pump 37 + 162 = 199 ft. The first pump is working with an average suction lift of 3 ft. which added to the 37 ft. discharge head makes a total of 40 ft.

Our test shows that the maximum capacity discharged is not any greater than the amount the first or slow speed pump will handle when operating alone, namely 155 G.P.M.

The second or high speed pump, which can handle up to 265 G.P.M., tries to pull more water through the first pump which is clearly shown on the Vacuum Pressure Gauge, which was connected to the pipe line leading from the discharge nozzle of the first pump into the suction nozzle of the second pump. The Pressure-Vacuum curve shows the gauge readings over the entire range, from shut-off where pressure is the same as the shut-off pressure of the pump alone, namely 15.7# = 37 ft. (the suction lift is 3.0 ft. making a total head of 40 ft.) to as much as 27" vacuum, or 30.5 ft. This is an interesting and entirely logical performance.

These results show that when connecting two centrifugal pumps, which have different maximum capacities, in series, the pressure of the large pump is reduced if the large pump has to pump through the small pump beyond its maximum capacity rating, and the capacity of the small pump controls the output when the small pump is ahead of the large pump.

Atmospheric Pressures, Equivalent Heads and Suction Lift

FIG. 33

	Atmospheric Pressure per Sq. In. in Pounds	Vacuum Gauge Reading in Inches of Mercury	Equivalent Head of Water in Feet (Theoretical Maximum)
Sea level	14.70	30.00	33.95
¼ mile (1,320 feet) above sea level	14.02	28.65	32.38
½ mile (2,640 feet) above sea level	13.33	27.25	30.79
¾ mile (3,960 feet) above sea level	12.66	25.87	29.24
1 mile (5,280 feet) above sea level	12.02	24.57	27.76
1¼ miles (6,600 feet) above sea level	11.42	23.34	26.38
1½ miles (7,920 feet) above sea level	10.88	22.24	25.13
2 miles (10,560 feet) above sea level	9.88	20.00	22.82

Vapor Pressures of Water at Various Temperatures

FIG. 34

Temp. Fahr.	Absolute Pressure of Vapor Lbs. per Sq. In.	Vacuum in Inches of Mercury	Theoretical Max. Depth of Suction in Feet	Temp. Fahr.	Absolute Pressure of Vapor Lbs. per Sq. In.	Vacuum in Inches of Mercury	Theoretical Max. Depth of Suction in Feet
79.0	.5	28.99	32.8	182.9	8	13.63	15.4
102.1	1	27.88	31.6	188.3	9	11.60	13.1
126.3	2	25.85	29.3	193.2	10	9.56	10.8
141.6	3	23.83	27.0	197.8	11	7.52	8.5
153.1	4	21.78	24.7	202.0	12	5.49	6.2
162.3	5	19.74	22.3	205.9	13	3.45	3.9
170.1	6	17.70	20.0	209.6	14	1.41	1.6
176.9	7	15.67	17.7	212.0	14.7	0.00	0.0

Temperature Conversion Table

FIG. 35

CENTIGRADE TO FAHRENHEIT
(ADD 32 TO TOTAL AFTER USING TABLE)

DEGREES CENTIGRADE

	0	1	2	3	4	5	6	7	8	9
0	0	1.8	3.6	5.4	7.2	9.0	10.8	12.6	14.4	16.2
1	18.0	19.8	21.6	23.4	25.2	27.0	28.8	30.6	32.4	34.2
2	36.0	37.8	39.6	41.4	43.2	45.0	46.8	48.6	50.4	52.2
3	54.0	55.8	57.6	59.4	61.2	63.0	64.8	66.6	68.4	70.2
4	72.0	73.8	75.6	77.4	79.2	81.0	82.8	84.6	86.4	88.2
5	90.0	91.8	93.6	95.4	97.2	99.0	100.8	102.6	104.4	106.2
6	108.0	109.8	111.6	113.4	115.2	117.0	118.8	120.6	122.4	124.2
7	126.0	127.8	129.6	131.4	133.2	135.0	136.8	138.6	140.4	142.2
8	144.0	145.8	147.6	149.4	151.2	153.0	154.8	156.6	158.4	160.2
9	162.0	163.8	165.6	167.4	169.2	171.0	172.8	174.6	176.4	178.2

FAHRENHEIT TO CENTIGRADE
(SUBTRACT 32 BEFORE USING TABLE)

DEGREES FAHRENHEIT

	0	1	2	3	4	5	6	7	8	9
0	.0	.5	1.1	1.6	2.2	2.7	3.3	3.8	4.4	5.0
1	5.5	6.1	6.6	7.2	7.7	8.3	8.8	9.4	10.0	10.5
2	11.1	11.6	12.2	12.7	13.3	13.8	14.4	15.0	15.5	16.1
3	16.6	17.2	17.7	18.3	18.8	19.4	20.0	20.5	21.1	21.6
4	22.2	22.7	28.3	23.8	24.4	25.0	25.5	26.1	26.6	27.2
5	27.7	28.3	28.8	29.4	30.0	30.5	31.1	31.6	32.2	32.7
6	33.3	33.8	34.4	35.0	35.5	36.1	36.6	37.2	37.7	38.3
7	38.8	39.4	40.0	40.5	41.1	41.6	42.2	42.7	43.3	43.8
8	44.4	45.0	45.5	46.1	46.6	47.2	47.7	48.3	48.8	49.4
9	50.0	50.5	51.1	51.6	52.2	52.7	53.3	53.8	54.4	55.0

THE DEMING COMPANY, Salem, Ohio, U. S. A.

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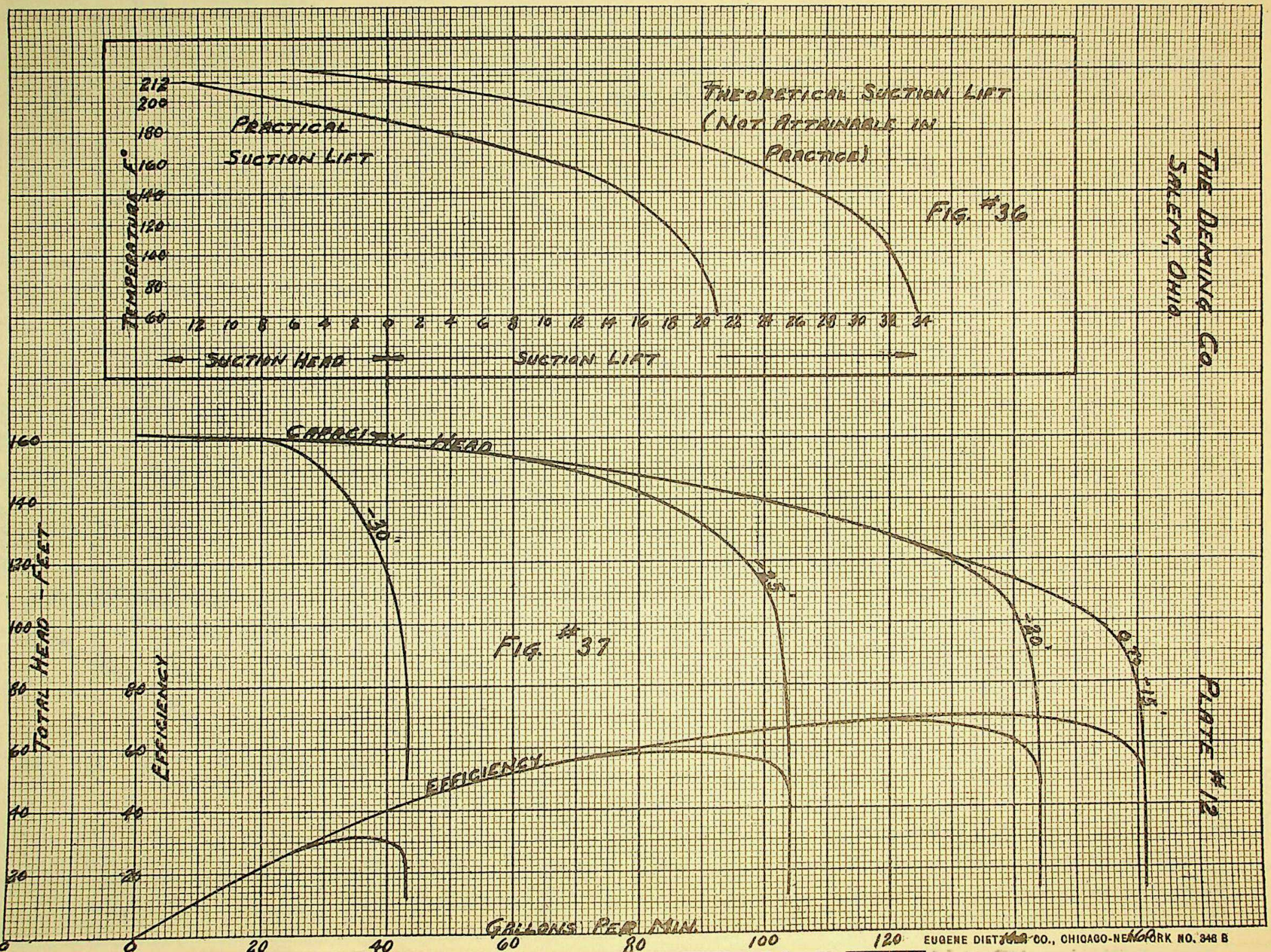

EUGENE DIETZGEN CO., CHICAGO-NEW YORK NO. 348 B

EUGENE DIETZGEN CO., CHICAGO-NEW YORK NO. 346 B

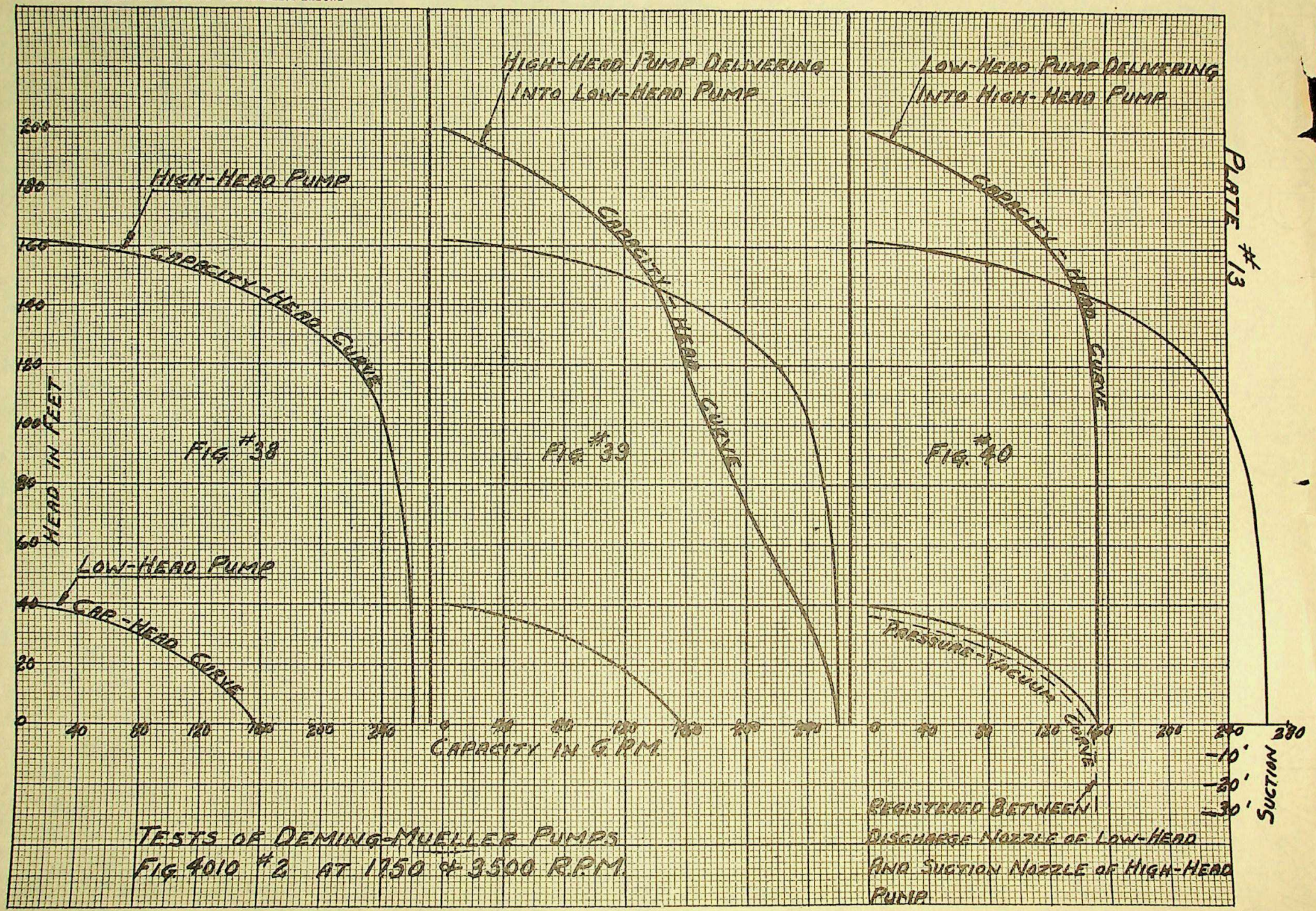

Plate #13

Tests of Deming-Mueller Pumps
Fig. 4010 #2 at 1750 & 3500 R.P.M.